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NON-DESTRUCTIVE TESTING BY GAMMA RAYS FROM RADIUM.*

BY ROBERT F. MEHL, PH.D.†

THE FIELD OF APPLICATION OF NON-DESTRUCTIVE TESTING.

In the present development of a highly mechanistic civilization the material engineer is constantly called upon to furnish materials of a quality higher than that of those previously used. In the first quarter century of its life, up to a very few years ago, the steel industry had its profit, and therefore its interest, practically exclusively in the production of steel in tonnage quantities. Naturally this interest will not lag, but recently there have been many manifestations of an increasing demand for quality, as rep-

* Published with permission of the Navy Department.

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resented by the multitude of new alloys which have been put on the market to meet extraordinarily stringent requirements. These are permitting engineers to design machinery and equipment far superior to that possible only a few years ago.

This demand for quality imposes one very definite requirement upon metallurgical production, even in the group which cannot be classed as that of special products, ordinary irons and steels: some method of inspection must be applied which will assure the engineer that the materials are in quality actually what he thinks them to be. In most cases this means a non-destructive method of testing, that is, one which will not destroy the piece as it tests it.

Many of the recently developed non-destructive testing methods are now common knowledge to the general public. X-ray methods of inspection have proved very satisfactory; the electric-magnetic method developed by Sperry for the detection of transverse fissures in rails has been an outstanding success. Besides these well-known methods, many electrical and magnetic methods have been developed including methods depending upon the change in electrical resistance with the presence of defects, the change in flux density, and other magnetic properties. Sonic methods, depending upon the resonating characteristics of a metal structure, have given promise of real value and will doubtless be greatly developed within the next few years. All of these methods aim to give the engineer some indication of the quality of the material which he wishes to use, especially with respect to the presence of physical discontinuities within the metal structure.

Radiography by X-rays is one of the oldest in the nondestructive testing field. It reveals a remarkably complete record of the interior state of the object radiographed and is being increasingly adopted by industry. Its particular field of application is to steel castings and welds, although it has also been used to study such problems as the alinement of elements in radio tubes, and the centering of the core in golf balls, and so on.

NOTE: The American Institute of Mining and Metallurgical Engineers has recently organized a committee on Non-Destructive Testing, of which the present writer is Secretary, the object of which is to collect all available information on the various non-destructive methods of test proposed and to be in a position to make trustworthy recommendations to the industry concerning these methods.

RADIOGRAPHIC NON-DESTRUCTIVE TESTING.

It may be said that X-ray radiography offers two types of service to industry; the first in demonstrating the defects in any or every piece of a manufactured article. Such mass inspection is rarely carried out, for its expense is very great. In the mass production of articles of identical design, industry has adopted the policy of inspecting only a fraction of the total output, well distributed according to principles of sampling. Such a program is now in force in one of America's largest manufacturing companies of pistons for automobiles. The second type of service is in the development of better technique of manufacture. This application is especially striking in the steel casting and welding field. In the hands of capable operators it has been possible by using radiographic methods to alter the casting technique in such a way that defects demonstrated by the radiographic inspection no longer occur. Similar success has attended the use of radiographic methods in the manufacture of welds.

The advantages in the X-ray method of radiography are now well known and need hardly be stressed; an intimate record of the internal characteristics of the radiographed piece is obtained, a record which, in the hands of a capable operator, may be adequately interpreted. The method suffers from some limitations which restricts the application of the method. The X-ray tube and necessary auxiliary equipment are rather cumbersome and are portable only in a very restricted sense, the equipment requires a rather experienced operator, and with the equipment at present available it is practically impossible to take radiographs through sections of steel greater than approximately 3 inches.

The interest of Captain E. G. Oberlin, then Assistant Director of the Naval Research Laboratory, in the application of X-rays to Naval problems led the present author to suggest that a research project be set up to determine what use might be made of gamma rays from radium for the detection of defects in metallurgical materials; this was undertaken in the summer of 1929 in collaboration with Dr. Charles S. Barrett of this Laboratory and Professor Gilbert E. Doan of Lehigh University, then on leave and employed at this Laboratory.

The interest in the gamma ray lay chiefly in its much shorter wave length. It was apparent from the beginning that if satisfactory radiographs could be taken with gamma rays a number of real advantages would accrue; the shorter wave length would permit taking pictures through thicknesses of steel far beyond that possible with X-rays, the source of radiation (radium held in a very small capsule) would be entirely portable, making it possible to radiograph structures in almost any conceivable location or position.

In general the work on radiography by gamma rays was successful. The present article will attempt to bring to the attention of the readers of this JOURNAL the results obtained, the necessary technique, and the general features characterizing the method.

RADIOGRAPHY BY GAMMA RAYS.

Physics of Radium—In order to make this account clear and complete it will be necessary to say something concerning the behavior of the radium which produces the gamma rays.

It is a characteristic of all radioactive elements, of which radium is the best known, to suffer an atomic decomposition, the elements transforming fundamentally to elements of different physical and chemical properties. During this decomposition, stepwise from element to element, various rays are emitted, among which is the gamma ray.

These decompositions may be represented by a series such as that given below:

$$\text{Ra} \xrightarrow[1580\text{ yrs.}]{\alpha} \text{Ra Eman.} \xrightarrow[3.85\text{ days}]{\alpha} \text{RaA} \xrightarrow[3.05\text{ min.}]{\alpha} \text{RaB} \xrightarrow[26.8\text{ min.}]{\beta,\gamma} \text{RaC} \xrightarrow[19.5\text{ min.}]{\beta,\gamma} \begin{cases} \text{RaC}' \xrightarrow[10^{-6}\text{ sec.}]{\alpha} \\ \text{RaC}'' \xrightarrow[1.32\text{ min.}]{\beta,\gamma} \end{cases} \text{RaD} \quad 1.32\text{ min.}$$

This series shows that radium transforms itself into an element known as radium emanation (known chemically as radon), which in turn decomposes into radium A, and so on to radium B, and C', etc. Each of these decompositions is attended by a characteristic emission of rays, of which there are three types: the alpha, the beta, and the gamma. The alpha ray is constituted of fast flying helium atoms with a double positive charge, the beta ray

of very fast flying electrons, and the gamma ray of light waves in no way fundamentally different from X-rays or ordinary light except in possessing an extremely short wave length. It may be seen from the above series that no gamma ray is emitted until radium B decomposes to radium C. It is believed that the chief part of the gamma ray emanation, however, occurs when radium C decomposes into its products.

It is an easy matter to stop the alpha and beta rays by absorption in thin films of metal. The gamma ray, however, is extremely penetrating, and as we shall see, will penetrate without great loss of intensity many inches of steel.

It will be noted in the above series that under each element is given a time period. This represents the time necessary for that particular radioactive element to decrease in strength to one-half the value obtaining initially. These elements do not lose their strength linearly with time but logarithmically, and accordingly it is quite impossible to give the full life of the elements. For this reason the convention of the one-half life period has been adopted. This relationship may be seen in Figure 1, which applies to radium emanation (radon). In approximately four days, as the diagram shows, the radiating strength of the radon has decreased to one-

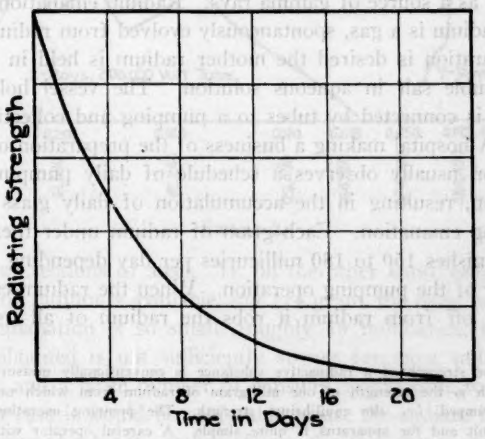

FIGURE 1.

half the initial strength. In another four days this one-half strength has again decreased by one-half to one-fourth of the original strength, and so on.

It may also be seen from this series that either radium or radium emanation may be used as a source of gamma rays, since the gamma ray does not appear until radium B is formed and since naturally the alpha radiation accompanying the transformation of radium to radon is of no value in radiography; nor is there any practical objection to using radium and radon interchangeably. In special cases one or the other may prove more suitable.

When radium itself is used as a source of gamma rays it is always held in sealed metal capsules, usually needle-like in form, and obtains in those capsules in the form of a radium salt, usually the insoluble radium sulphate. For all practical purposes the radiating life of such a radium capsule is infinite, for the half life period of radium is about 1600 years. The use of radium salt directly for radiography has the advantage that there is no attention required by the source once it has been suitably prepared and that its radiating strength may be considered for practical purposes as constant with time.

The medical profession makes some use of radium in this form. It is, however, possibly more customary to use radium emanation (radon) as a source of gamma rays. Radium emanation (radon) unlike radium is a gas, spontaneously evolved from radium. When its preparation is desired the mother radium is held in the form of a soluble salt in aqueous solution. The vessel holding this solution is connected by tubes to a pumping and collecting apparatus. A hospital making a business of the preparation of radium emanation usually observes a schedule of daily pumping of the emanation, resulting in the accumulation of daily glass capsules containing emanation. Each gram of radium under these conditions furnishes 150 to 180 millicuries per day depending upon the efficiency of the pumping operation. When the radium emanation is drawn off from radium it robs the radium of all its gamma

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NOTE: The strength of a radioactive substance is conventionally measured in millicuries, which is the strength of one milligram of radium from which no emanation has been pumped, *i.e.*, the equilibrium strength. The pumping operation is by no means difficult and the apparatus is quite simple. A careful operator with but brief experience can attain an efficiency from 90 to 95 per cent.

ray activity for the moment until the radium itself spontaneously generates new emanation. In other words, the radioactivity of the radium itself is being transferred to the series of capsules. The total strength of the capsules gathered over a period of a month or so should then equal the strength of the radium from which no emanation had been pumped; because of the factor of efficiency this is never quite attained.

The volume occupied by radium emanation is very small, 1000 millicuries occupying only 0.6 cubic millimeters. Accordingly, the gas can be concentrated in a very small capsule which is often advantageous for purposes of radiography.

The requirement of point source for good definition in radiography, about which more will be said later, has its bearing upon whether radium or radium emanation should be used. If a large amount of radium is available, say several grams, one day's product in radium emanation pumped into a small capsule will be powerful enough to do most radiographic tasks, even including

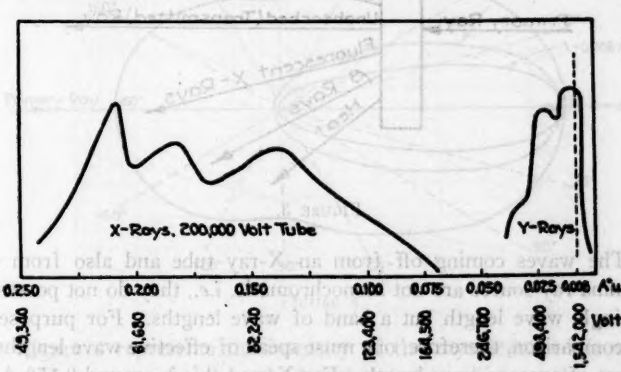

FIGURE 2.

very heavy sections of steel. If, on the other hand, only a small quantity of radium is available, say 1/4 gram, the daily product in radium emanation is so small, roughly 45 millicuries, that each capsule obtained is not sufficiently strong for most radiographic purposes, and a large number of capsules would have to be used. In such a case it would be preferable to use the radium itself, for the grouping of a large number of capsules as a gamma ray source

is in general not advisable because of the loss of the point source characteristics. It is quite obvious that the type of source to use, either radium or radium emanation, will depend upon practical details in the operation of the method in any particular organization.

As already stated the most important characteristic of gamma rays from a radiographic viewpoint is their much shorter wavelength. It is of interest in this connection to compare wave length of gamma rays with that of X-rays obtained from the high voltage tube now used industrially. These wave lengths are shown in Figure 2.

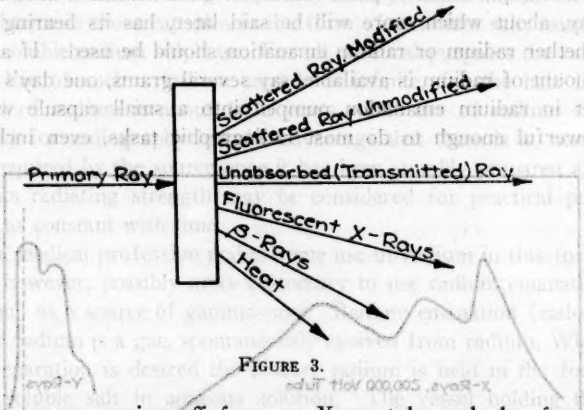

FIGURE 3.

The waves coming off from an X-ray tube and also from a gamma ray source are not monochromatic, *i.e.*, they do not possess a single wave length but a band of wave lengths. For purposes of comparison, therefore, one must speak of effective wave lengths, a sort of average wave length. For X-rays this is around 0.150 A; for gamma rays about 0.008 A. As we shall see, this much shorter wave length has important consequences for radiography.

Since radiography by gamma rays depends upon the passage of gamma rays through matter and their recording upon a photographic film, it might be well to see what happens when such a ray strikes a piece of metal.

Figure 3 shows the changes suffered by gamma rays (or by X-rays) in passing through matter. The beta rays and the heat

rays are of no importance to us since the beta rays are absorbed by the film holder and the heat has no photographic action. The fluorescent X-rays are of scientific interest but are so weak as to be unimportant in radiography. The three types of rays of real importance to radiography are the unabsorbed (transmitted) ray and the two types of scattered ray. The unabsorbed, transmitted, ray is the chief ray. Its importance in radiography does not need stressing. The scattered rays, however, deserve careful consideration, for generally they operate to decrease the definition obtained in a radiographic film, producing the familiar haloes and fogging characteristics of films taken by X-rays without suitable lead screening to prevent their harmful action. One advantage in gamma rays lies in the fact that their scattering is inherently less disadvantageous.

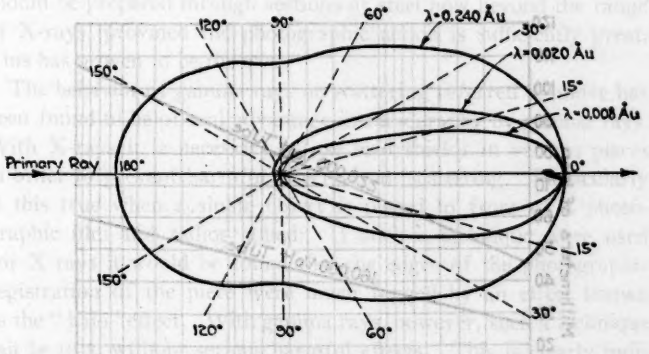

FIGURE 4.

Figure 4 represents the intensity of the rays scattered from a central point, as shown by the lengths of the radii from this point. It may be seen that with the shorter wave lengths the scattering is almost entirely ahead, whereas longer wave lengths give considerable scattering which results in a general film fogging and the production of haloes. Little need be said about the modified rays. This refers simply to a change in wave length which the rays suffer upon scattering. The effect will be only slight for X-rays

Å signifies an Ångström unit which is 10^{-8} cm. or 0.00000001 cm.

but quite profound for gamma rays. Since the modification consists in an increase in wave length, this increase in wave length will result in a greater absorption of the scattered rays and will thus tend to decrease the scattering effect indicated in the diagram.

The essential difference between X-ray radiography and gamma ray radiography, therefore, resides in the much shorter wave length of the gamma ray. Let us see what the consequences of this shorter wave length are, first with respect to the photographic action.

The exposure curves to obtain satisfactory radiation through steel and other metals by X-rays have been prepared by a number of investigators. The diagram here given, Figure 5, shows such a carefully determined curve.

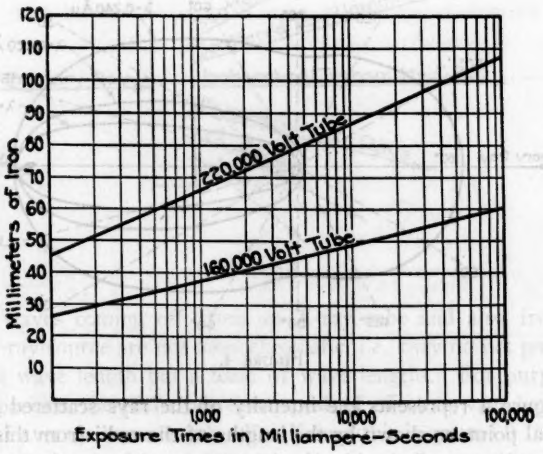

FIGURE 5.

In order to obtain a comparison between the X-ray and the gamma ray from the exposure-time standpoint, let us consider the ratios in time between that necessary to prepare a photograph through 6 inches of steel and that through 3 inches of steel first with X-rays and then with gamma rays.

Reading from the exposure curve for X-rays this ratio is

$$\left(\frac{t_6''}{t_3''}\right)_{\text{x rays}} = 5000$$

In the lack of an exposure curve for gamma rays this ratio may be calculated theoretically from the absorption coefficients, for the ratio should stand in direct proportion to these coefficients. Such a calculation indicates a ratio

$$\left(\frac{t_6''}{t_3''}\right)_{\text{gamma rays}} = 45$$

which can and has been confirmed experimentally.

These ratios show that the increased time to obtain a radiograph through additional thickness for gamma rays is very much less than that for X-rays and indicate also that satisfactory radiographs should be prepared through sections of steel now beyond the range of X-rays, provided the photographic action is sufficiently great. This has proven to be the case.

The behavior of gamma rays on scattering referred to above has been found to be of real advantage in radiography by gamma rays. With X-rays it is necessary to use lead shields in various places in order to prevent harmful effects from scattering. Particularly is this true when a single object is placed in front of a photographic film and radiographed. If such a technique were used for X-rays it would be found that the edges of the photographic registration of the piece were much fogged by an effect known as the " halo" effect. With gamma rays, however, such a technique can be used without serious harmful effects. This is clearly indicated on the radiograph of a bronze valve casting shown in the next two illustrations (Figures 6 and 7). The photograph (Figure 6) shows the casting which was machined on the ports. The radiograph following (Figure 7) shows a complete radiograph. It will be noted that there is no halo effect around the edges of the piece and it will be further noticed that the registration of the internal threads is quite distinct and entirely suitable for the purposes of radiography. Except for one very small defect on the lower port, the bronze valve casting proved to be sound.

The use of the shorter wave length for radiography has one feature which may not be immediately apparent. When a very

irregular piece is to be radiographed it is always a puzzle to know whether to choose an exposure time correct for the thin sections or for the thick. With X-rays this is a very serious problem, for an exposure time suitable for the thick sections will inevitably badly over-expose the thin sections; whereas if the exposure time is chosen correctly for the thin sections the thick sections will be badly under-exposed. It is a direct physical consequence of the shorter wave lengths of gamma rays that this choice of exposure times is not so important, *i.e.*, it is easily possible to pick an exposure time which will give correct exposure for both thick and thin sections on the same radiographic film. This will be evident from the radiographs of the casting with sand inclusions to be presented later.

PRACTICAL DETAILS IN GAMMA RAY RADIOGRAPHY.

As we have stated in the foregoing, either radium or radium emanation can be used, depending upon practical considerations in the organization interested in using it. Whether radium or radon is used the radioactive source will always be in the form of a small capsule roughly the size of a 22 caliber cartridge. In order to absorb the alpha and beta rays, which are useless from the radiographic standpoint and harmful physiologically, the capsule is always enclosed with a thin sheath of silver or platinum or brass which is capable of absorbing these rays. Generally a millimeter thickness is more than sufficient to let through only the gamma rays.

The advantage of a source of small dimensions may be seen upon a little reflection. A large source will produce a ray coming from a multitude of separate points, striking the film at differing angles. A defect in the piece some distance from the film will then be registered upon the film as a series of spots merging into one spot of confused outline. A point source, however, will, like an arc light, produce a sharp shadow.

The necessary apparatus need be only this small radioactive source and a film holder containing the films. Both the source and the films naturally are completely portable and may be taken anywhere. In fact, it is by no means a wild idea that a diver should take this equipment, suitably water-proofed, and attach

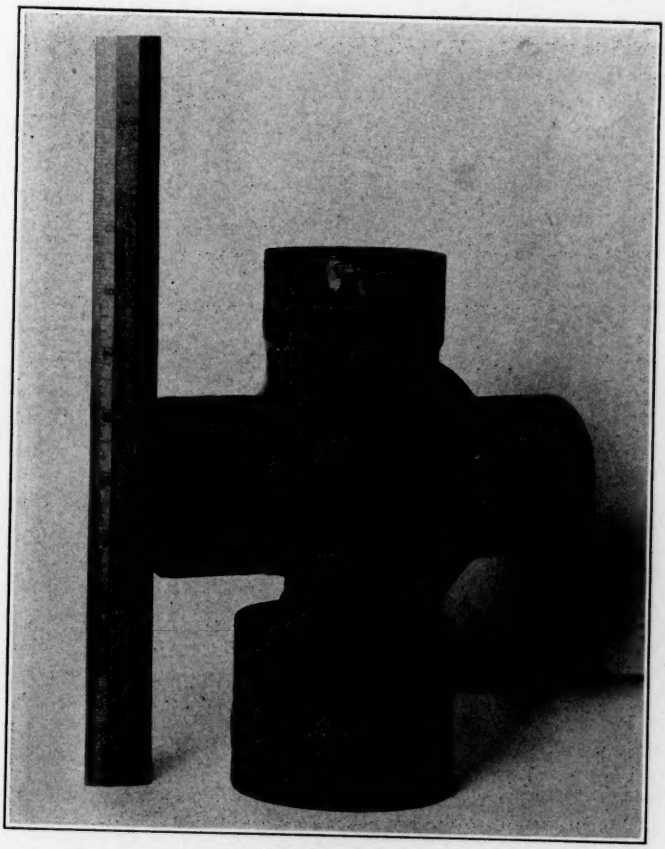

FIGURE 6.

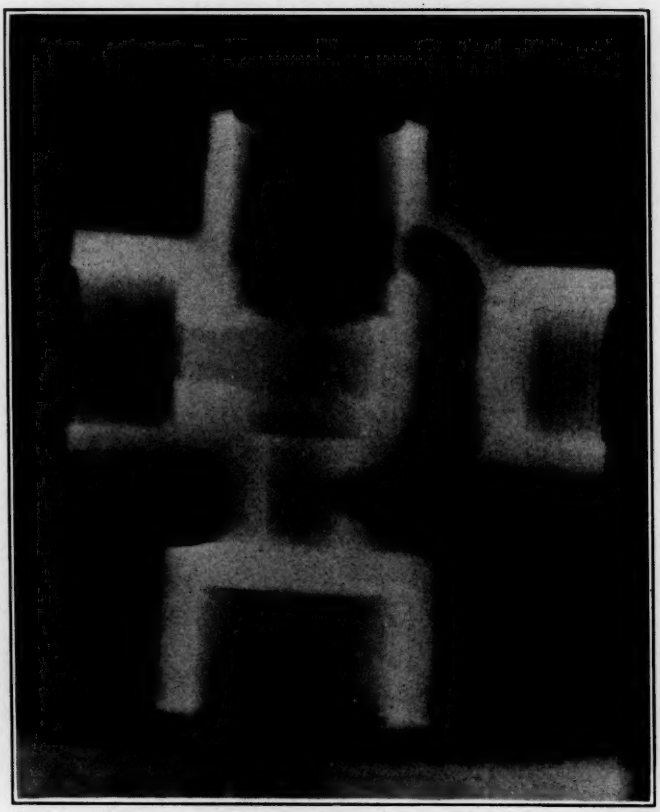

FIGURE 7.

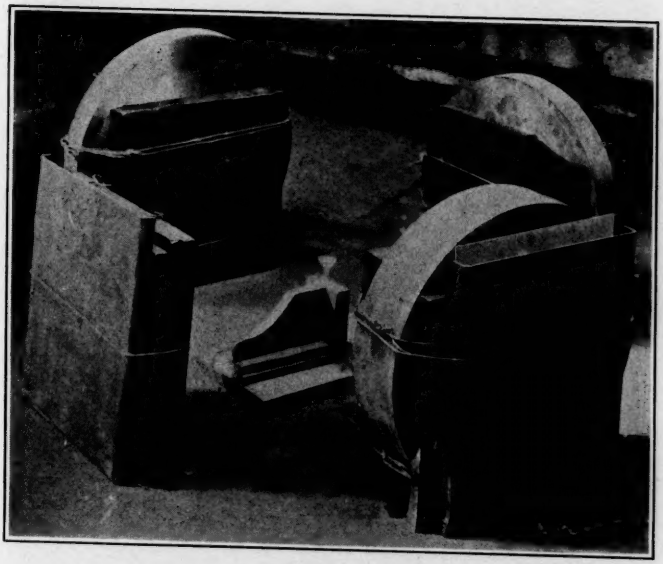

FIGURE 8.

FIGURE 9.

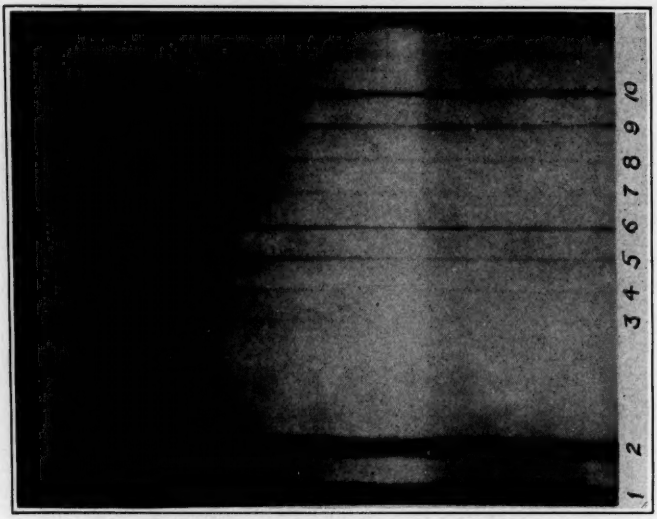

FIGURE 11.

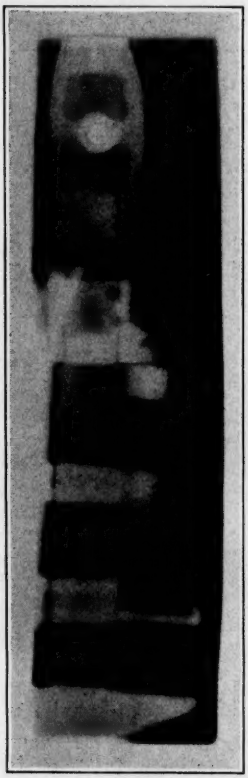

FIGURE 12.

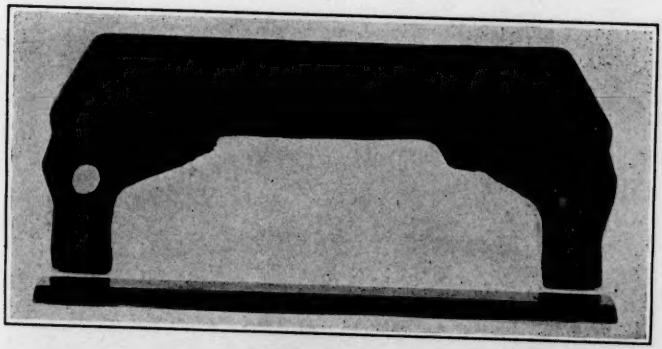

FIGURE 13.

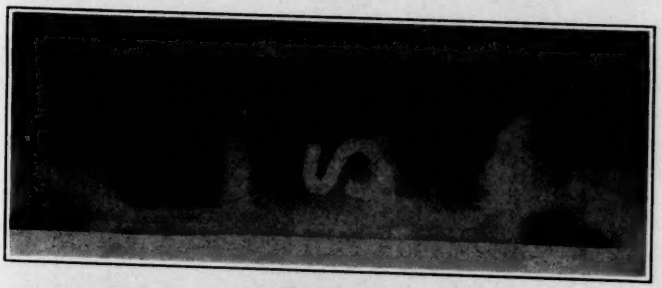

FIGURE 14.

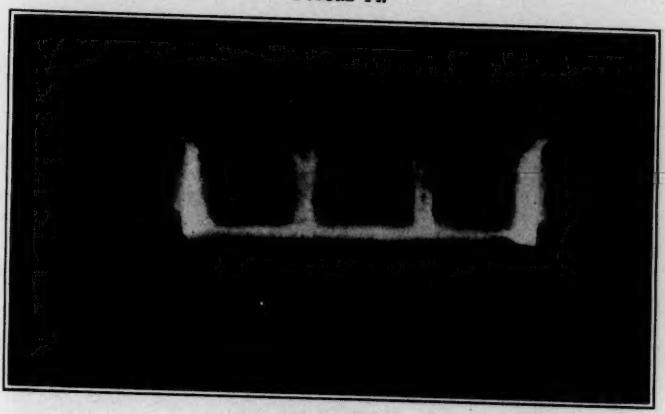

FIGURE 15.

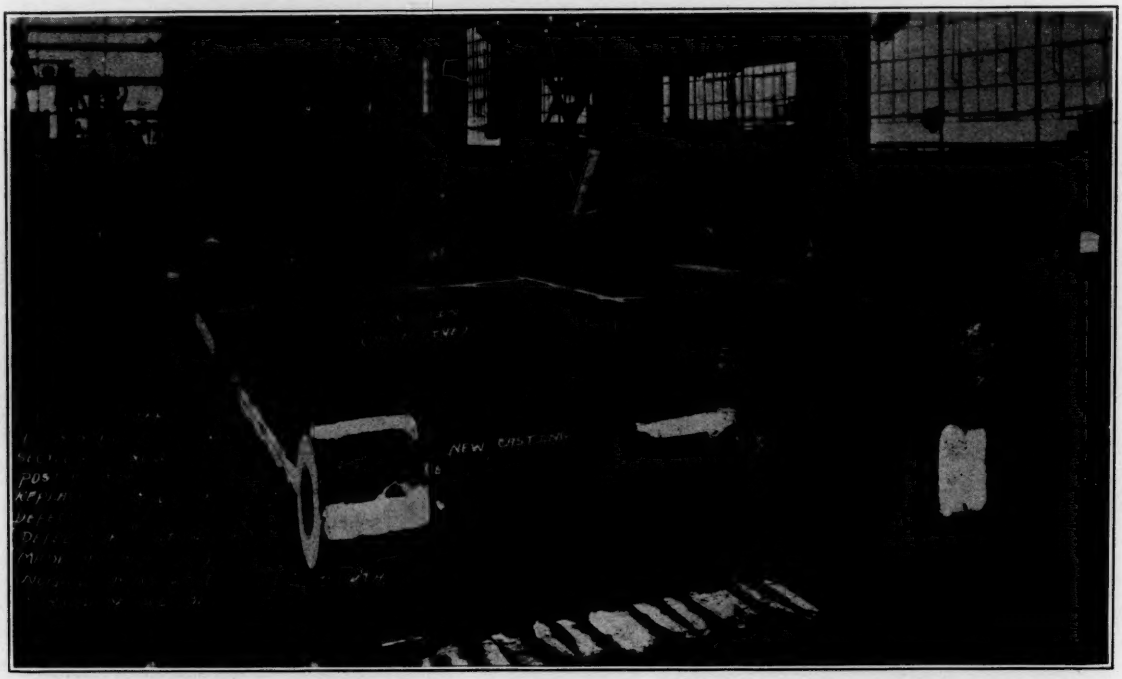

FIGURE 17.

it to a section of a ship under water, leaving it there for a suitable exposure time and then remove it, thereby obtaining a radiographic record of the section of interest without dry-docking the ship for the purpose. In general, this portability is of great value. It means that pieces to be radiographed need not be brought to the radiographic apparatus but that the radiographic apparatus can, in every case, be brought to the piece; so that however remote the piece may be from the laboratory or however awkward the piece may be for the purposes of radiography, the gamma ray method is superior from this standpoint to the X-ray method.

It must be remembered that radium radiates gamma rays in all directions, spherically. This means, again in contra-distinction to the X-ray method (which uses effectively only a beam), that the radioactive source may be surrounded as completely as possible with objects to be radiographed. Figure 8 shows the simultaneous radiography of four steel objects. It needs no extensive play of imagination to see that this number could be increased and that various tricks could be used to employ a large fraction of the total radiation issuing from the bulb. The opportunity for simultaneous radiography is also illustrated in Figure 9. In this particular case the object is a cast steel gun slide. The radium was placed in a small holder mounted on a tripod just visible on the inside of the casting. The wall of the casting in this case was four inches thick. The illustration shows an operator placing two films on the outside of the casting. Because of the spherical radiation of the gamma rays, at least two bands of 10 $\times$ 12 films could have been placed completely around this cylindrical casting and some fifty films exposed at one time. This opportunity for simultaneous radiography must always be kept in mind when the exposure times necessary for the production of suitable radiographs are considered. Obviously, with fifty photographs taken at one time on the cast steel gun slide, the time spent on each photograph should be taken as one-fiftieth of the exposure time noted.

The technique necessary for the preparation of radiographs with gamma rays is very similar to that for X-rays, except, as stated before, protection from scattering is not so important. The first information that is required before radiographs can be pre-

pared is the exposure time necessary to obtain suitable radiographs through a certain thickness of steel at a given distance with a radioactive source of known strength; in other words, an exposure curve is needed. Such an exposure curve has now been experimentally determined at this Laboratory and is reproduced in Figure 10.

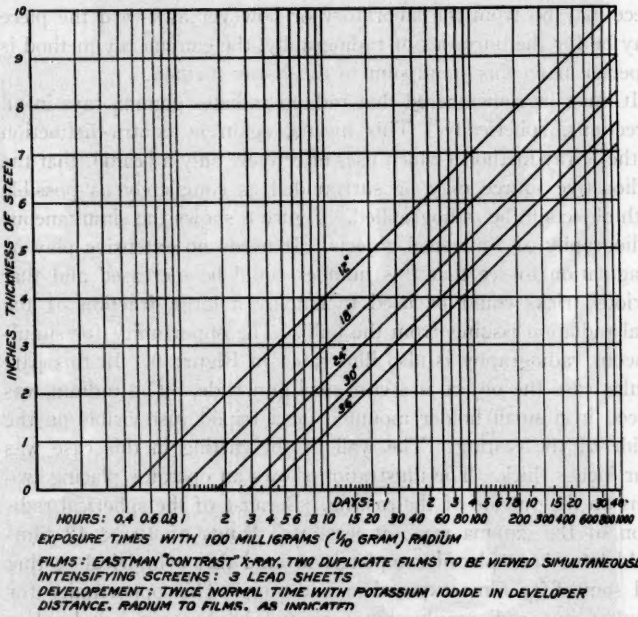

FIGURE 10.

It shows on semi-logarithmic paper the relationship between the exposure of the piece (in so many milligram hours) as related to the thickness of steel radiographed for different distances from the source to the film. It has been found that the exposure times can be shortened in a number of ways. First, by the use of duplicate films, which permits of shorter exposure times because the faint films obtained thereby, when viewed in duplicate, bring out the characteristics of the piece satisfactorily. In order

further to shorten the exposure time it has been found advisable to use three thin sheets of lead enclosing the two duplicate films. This lead acts as an enhancing screen, speeding up the photographic action of the gamma rays and therefore acts similarly to the Patterson screens much in use in the medical profession.

The exposure curve given therefore represents the exposure times against thicknesses of steel for the use of duplicate films and the use of three lead sheets as enhancing screens.

The distance from source to film is a matter of some concern in radiography. In general, the best conditions will be obtained at a great distance from source to film since the normal divergence of the beam will be less at large distances than at small and therefore the shadow registration of the defects will be more nearly like the defects themselves. However, the time of exposure increases very rapidly with distance and some compromise must be reached. It is difficult to give a distance which in general is best, since this will vary according to the characteristics of the piece radiographed. It is a matter rather largely of judgment in each particular application. The exposure curves also show what changes in exposure time must be made for changing distances. It is simply a representation of the inverse square law, *i.e.*, that the intensity of the beam falling upon an area varies inversely with the square of the distance. It is possibly needless to remember in connection with the exposure curve that the times of exposure, expressed in millicuries or milligram hours, may be attained by using large amounts of radium and smaller times, or small amounts of radium and longer times. As a matter of convenience, it frequently turns out that an overnight exposure is quite convenient and in order to ob-

NOTE: Patterson screens could also be used for our purposes but they are rather expensive and we have found lead to be in general quite satisfactory.

NOTE: The position and course of this exposure curve is dependent upon the speed of the reaction of the photographic film with the gamma ray. The curve given applies to Eastman's Contrast Duplitized Safety X-Ray films. A faster film would decrease these exposure times, and some results on new films now indicate a reduction of at least 20-25 per cent. Such a photographic advance is naturally of the greatest importance to gamma ray radiography, for the exposure times can then be decreased or the necessary quantities of radium will be less. Advances in the art and science of gamma ray radiography in the future will almost certainly be in photographic materials and processes.

tain the necessary data for an overnight exposure the following table was developed. This shows the amounts of radium or

TABLE 4—Strength of Radium in Milligrams and Initial Strength of Emanation in Millicuries Required for an Overnight Exposure (15 Hours) at the Distances and through the Thickness Indicated, with Two Lead Screens, Dupli-Tized Superspeed X-Ray Films, Over-development, and Film Density of 2:20.

Thickness of	Source to film, distance in inches							
	12		18		24		30	
iron in inches	Ra	Em	Ra	Em	Ra	Em	Ra	Em
1	5.3	5.6	12.0	12.7	21.3	22.5	33.3	35.2
2	12.6	13.3	28.3	29.9	50.3	53.1	78.6	83.0
3	29.3	31.0	66.0	69.7	117.3	123.8	183.3	193.6
4	63.0	66.6	141.6	149.6	251.6	265.8	393.1	415.3
5	132.5	140.0	298.2	315.0	530.0	560.0	828.1	875.0
6	296.5	313	666.6	704	1185	1252	1852	1956

radium emanation which must be used at the stated distances in order to get satisfactory radiographs through the thicknesses of steel indicated. It is merely a convenient way of reporting the data which may be obtained directly from the exposure curve given above.

The developing of the films is very little different from the usual X-ray technique except that the films are over-developed with potassium iodide in the developer in order to diminish fogging. Naturally, it is necessary to use light-tight envelopes to hold the film in place during radiography and to be careful about accidental fogging during the manipulation of the films in the darkroom.

RESULTS OBTAINED IN GAMMA RAY RADIOGRAPHY.

One of the first questions asked of any method of this sort is its sensitivity. In order to obtain information on this point an experiment was performed which involved the radiography of six steel plates, with a total thickness of four inches, in one of which plates was machined a series of slots of varying depths and widths. The radiograph, Figure 11, shows the radiographic registrations of these slots.

Slot 3 was but 2 mm. deep and 1 mm. wide and therefore of a depth only 2 per cent of the total thickness. This slot can be

very clearly seen on the original radiographs though the reproduction here hardly shows it. Slot 4 is very distinct and was 3 per cent of the total thickness. The lower part of this radiograph is much darker than the upper because the upper part had a lead sheath in the rear which prevented scattered rays from the rear striking the film. Although prevention of scattering in general for radiography is not highly important it is, nevertheless, of some advantage to decrease the resultant fogging as this photograph shows. The difference in the relative darkness is rather greater on this illustration than upon the original film.

It may therefore be concluded from this experiment and from many others performed since that the gamma ray method is quite sensitive to the detection of very thin defects of the order of 2 per cent (or less) of the total thickness. Such a sensitivity ordinarily is entirely satisfactory for radiographic inspection.

The details which can be obtained in this kind of radiograph may be seen in Figure 12 which is of a wrench. The original wrench was some 8–1/2 inches long, and of the ordinary proportions.

The next three illustrations are of considerable interest in connection with the X-ray and the gamma ray methods of radiography. The first photograph, Figure 13, shows the picture of an irregular steel casting some 12 inches long. At the ribs the total thickness was 1–1/4 inches. Between the ribs the thickness was but 1/4 inch. The first radiograph, Figure 14, shows an X-ray radiograph of this casting. It may be seen that a number of serious defects are registered with excellent precision. It may also be seen that the end sections and the ribs are badly underexposed, owing to the considerable difference in cross-section. The next radiograph, Figure 15, is a gamma ray radiograph of the same piece. The defects which may be discovered in the preceding radiograph are visible here also. It will be noted immediately that the contrast, however, is less in the gamma ray radiograph than in the X-ray radiograph. This is, as before stated, an inevitable consequence of the shorter wave length. The correlative

NOTE: In the following radiographs no exposure times will be listed since these may be simply read from the exposure curve given above as applying to the thickness concerned.

consequence, however, that of less sensitivity to changing cross-sections, is likewise evident in these radiographs, for it will be seen in the gamma ray radiograph that the end sections and the ribs are more nearly suitably exposed than in the X-ray radiograph. Nor must it be concluded that the diminished contrast in the gamma ray radiograph is a really serious disadvantage, for no defect could be found in the X-ray radiograph of this piece which could not independently be found in the gamma ray radiograph. The gamma ray radiographs are simply flatter in appearance and the eye must be accustomed to a diminished contrast. With this adjustment of eye, it is just as possible to detect small defects with gamma ray radiography as with X-rays.

The radiographs shown so far are a selected few from those prepared during the experimental development of this method. The method had its first important application to a sternpost casting aboard the U. S. S. *Chester.* This was a 9 ton casting in the shape of a shell, coming just forward of the rudder and bearing at the bottom the bending moment of the rudder. It is illustrated in the diagram (Figure 16) and in the following photograph (Figure 17) showing also the new casting made to replace the defective one.

After the installation of the "defective casting" upon the *Chester,* two cracks developed on the port side near the top. These cracks were promptly chipped out and welded, but considerable uncertainty remained concerning the quality of the rest of the casting. When the *Chester* put into dry-dock at Norfolk, representatives of this Laboratory went to Norfolk and took 24 radiographs of both the port and starboard sides of the casting.

The section of steel studied was only 1–1/2 inch thick and therefore the exposure times and the quantity of radium necessary were not great. The general result of this work was a demonstration that the metal on the port side was very poor; that the casting was full of shrinkage cracks; and that continued use of the casting would be dangerous.

The types of radiographs obtained are shown in the next few illustrations.

The radiograph in Figure 18 shows the presence in one section of the port side of the casting of numerous defects including cracks

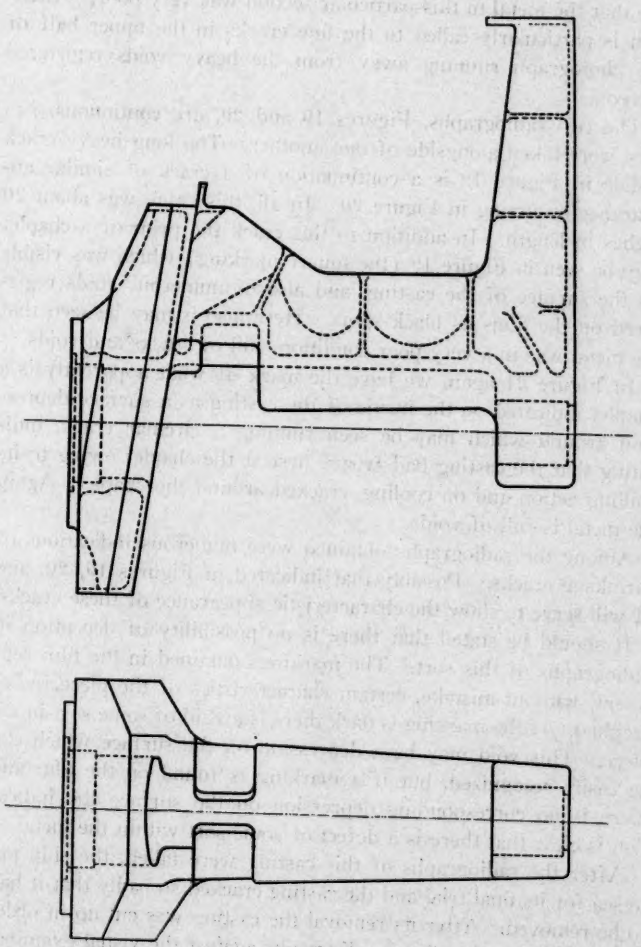

FIGURE 16.

and voids of various descriptions. It is clear to an experienced eye that the metal in this particular section was very poor. Attention is particularly called to the fine cracks in the upper half of the photograph running away from the heavy voids registered thereon.

The two radiographs, Figures 19 and 20, are continuous, *i.e.*, they were taken alongside of one another. The long heavy crack visible in Figure 19 is a continuation of a crack of similar appearance occurring in Figure 20. In all, this crack was about 20 inches in length. In addition to this crack the print of a chaplet may be seen in Figure 19 (the square marking) which was visible on the surface of the casting, and also of numerous voids registered on the films as black spots. Here also it may be seen that the metal was in a very poor condition, full of cracks and voids.

In Figure 21 again we have the mark of what apparently is a chaplet indicated on the inside of the casting as a surface depression around which may be seen running a circular crack, indicating that the casting had frozen first at the chaplet owing to its chilling action and on cooling, cracked around the chaplet. Again the metal is full of voids.

Among the radiographs obtained were numerous indications of shrinkage cracks. Possibly that indicated in Figures 19, 20, and 21 will serve to show the characteristic appearance of these cracks.

It should be stated that there is no possibility of deception in radiographs of this sort. The markings obtained in the film represent, without mistake, certain characteristics of the piece radiographed. If the marking is dark there is a void of some sort in the piece. This void may be a depression on the surface which can be easily recognized, but if a marking is found on the film and there is no corresponding depression on the surface the indication is clear that there is a defect of some sort within the metal.

After the radiographs of this casting were taken, the ship put to sea for its final trial and the casting cracked so badly that it had to be removed. After its removal the casting was cut up in order to check the results of the radiographs against the visual examination of the sectioned parts. In each case the indication on the radiograph was confirmed and it was found that the radiographs indicated in certain cases very fine cracks, showing that the sensi-

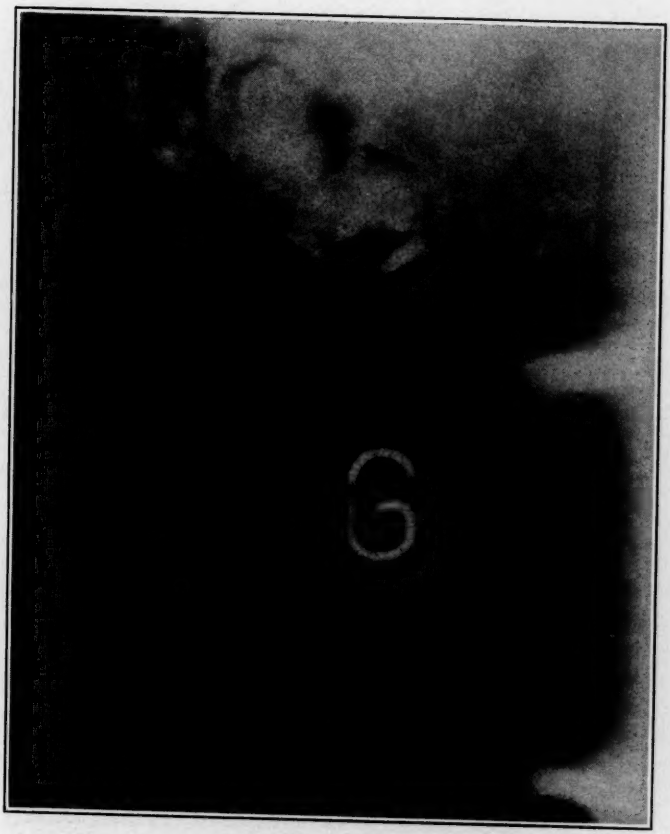

FIGURE 18.

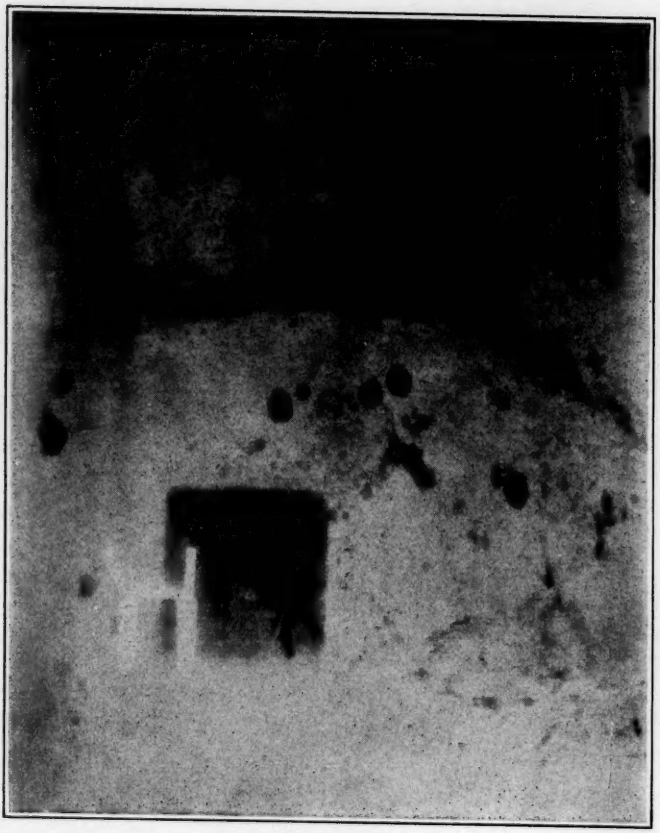

FIGURE 19.

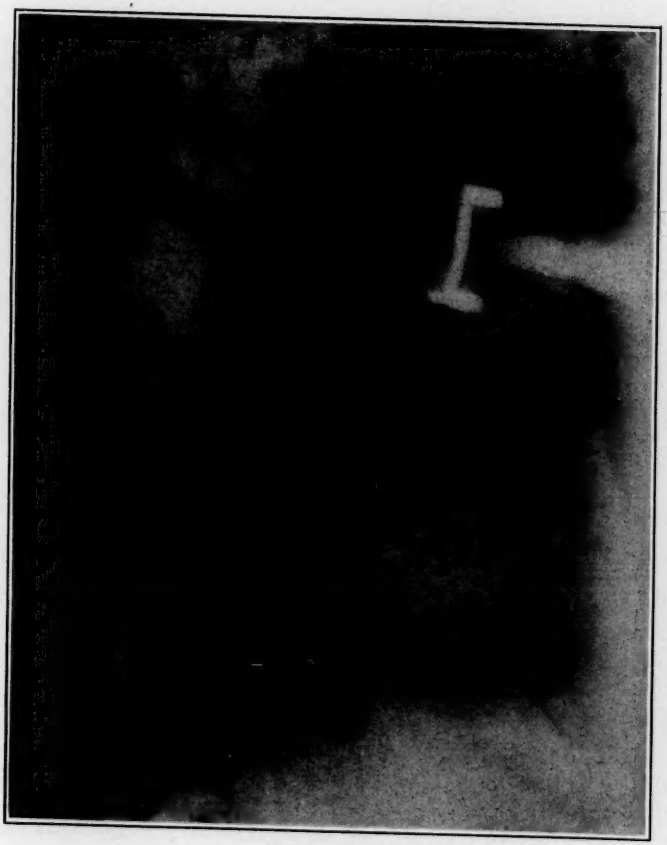

FIGURE 20.

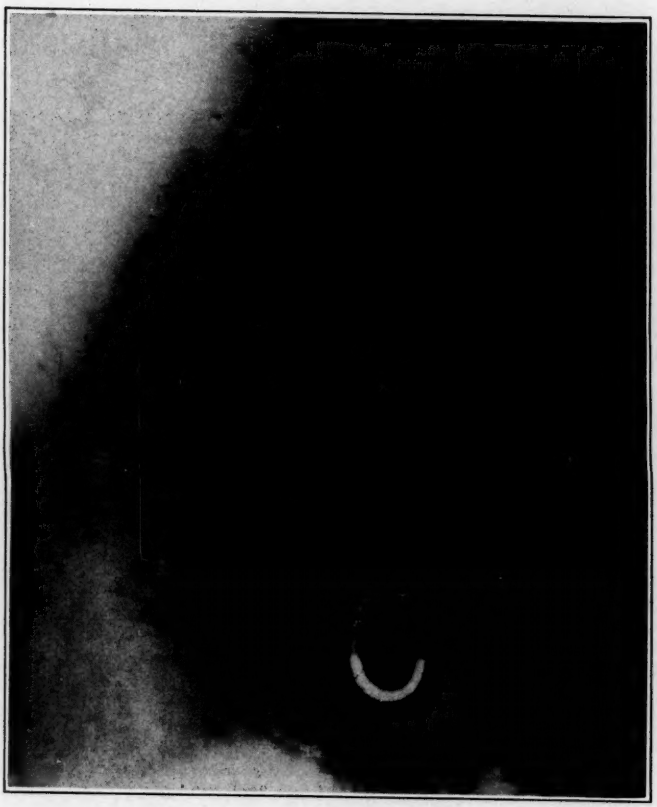

FIGURE 21.

FIGURE 22.

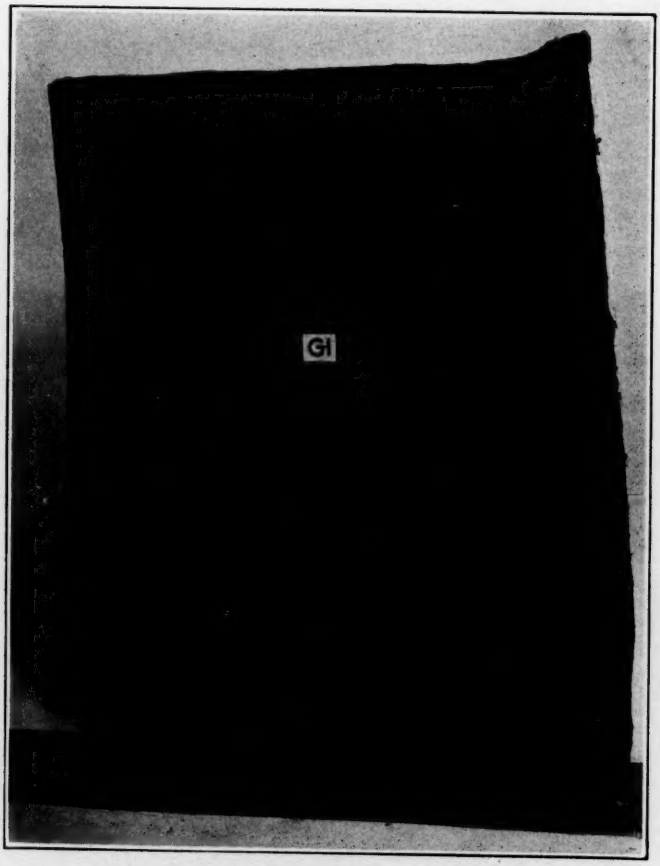

FIGURE 23.

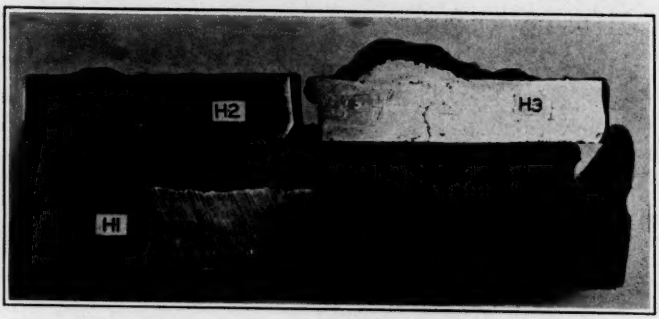

FIGURE 24.

FIGURE 25.

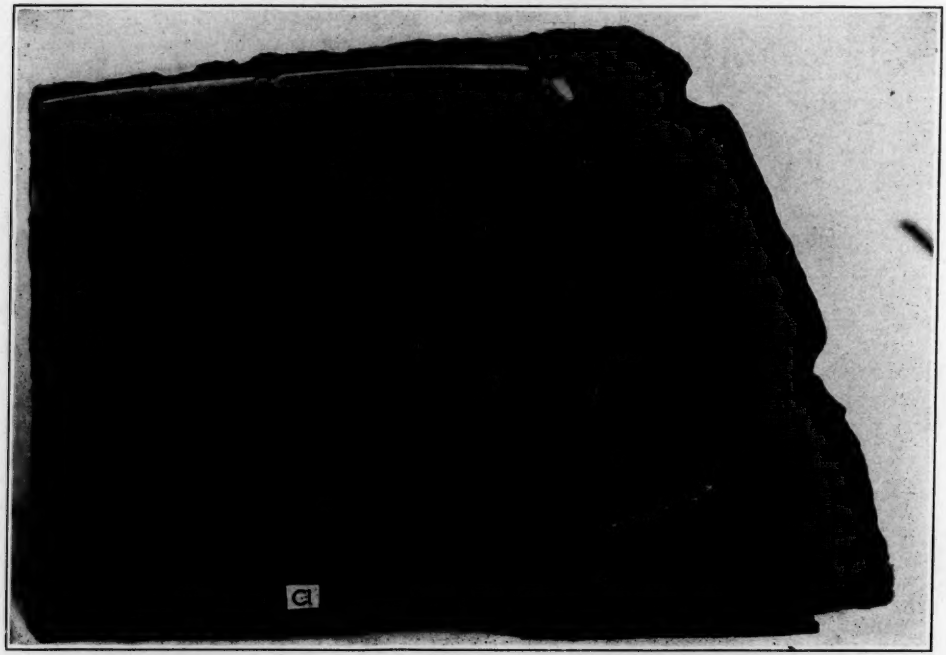

FIGURE 26.

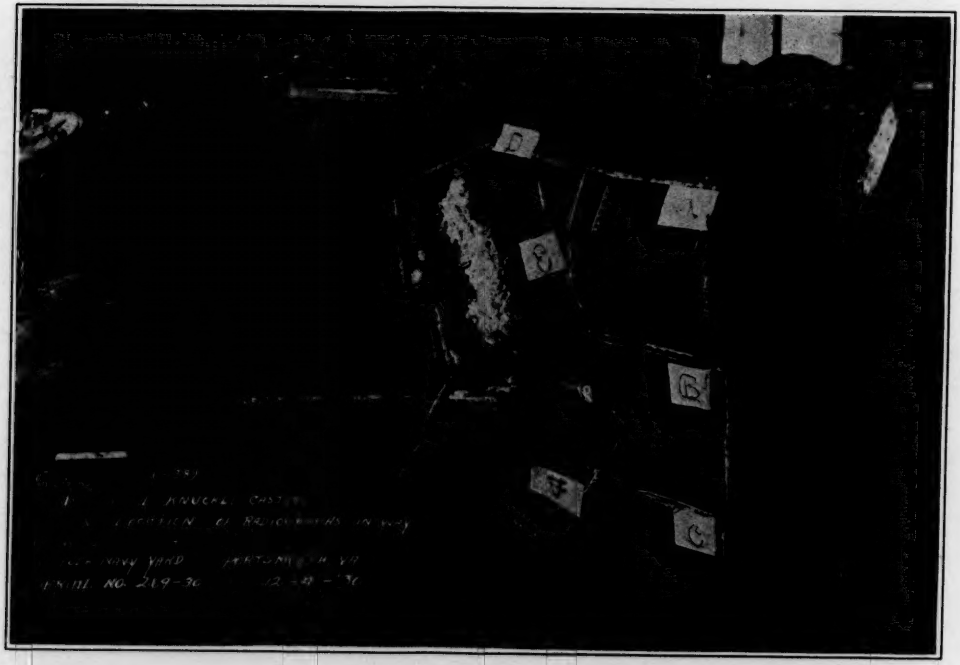

FIGURE 27.

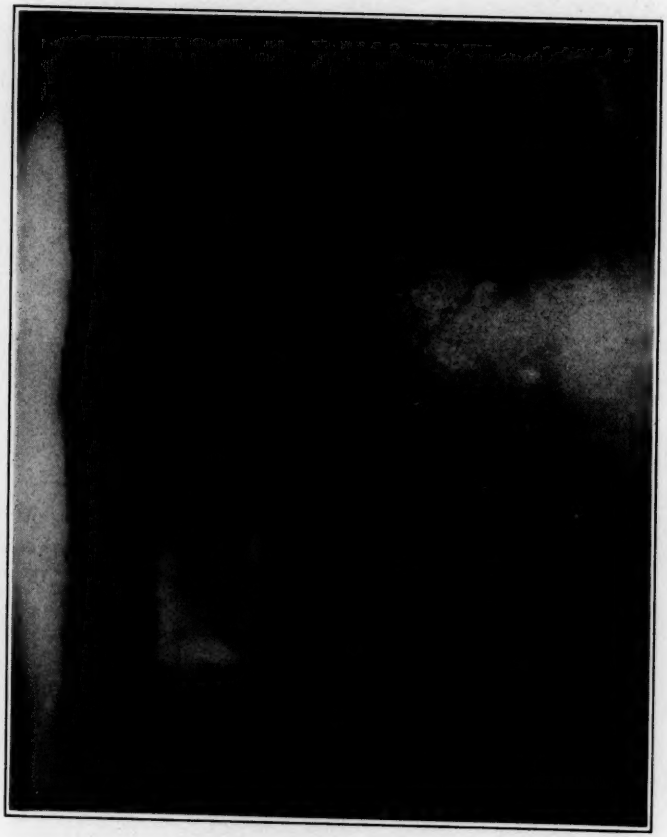

FIGURE 28.

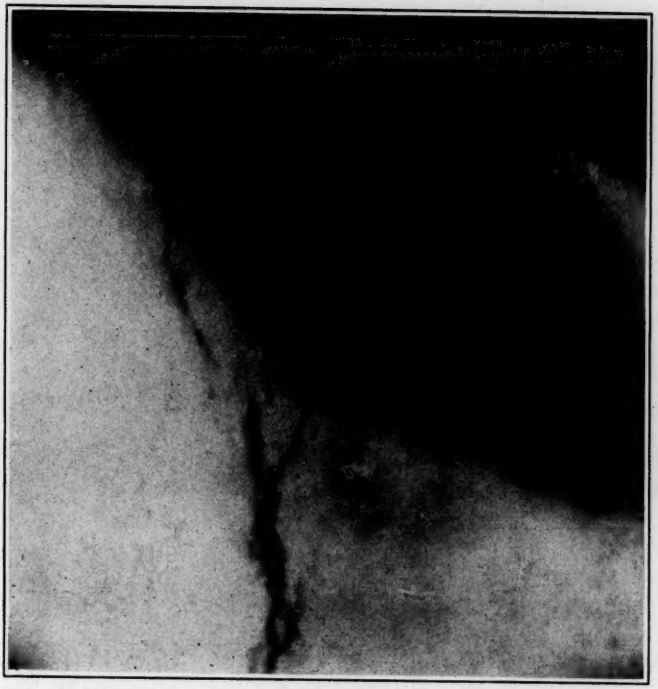

FIGURE 29.

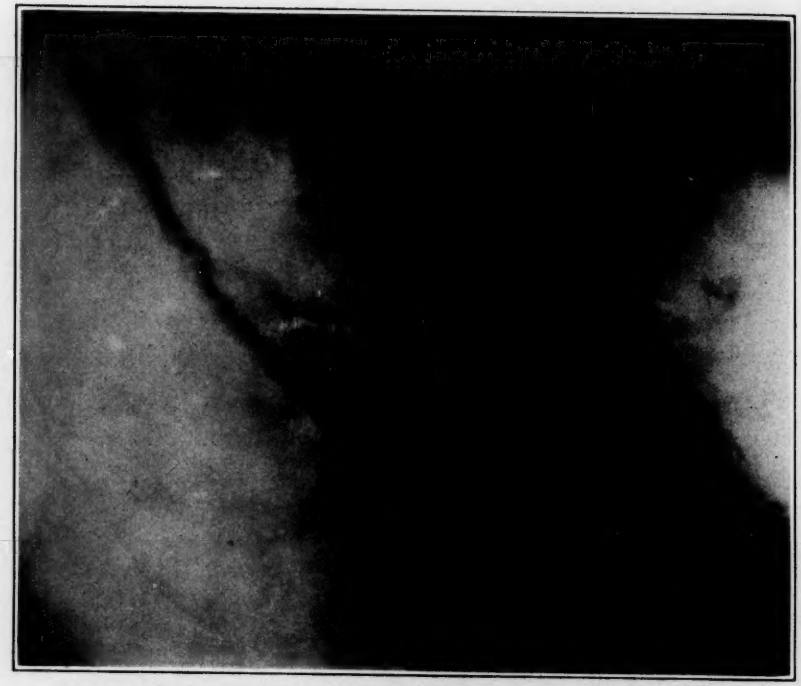

FIGURE 30.

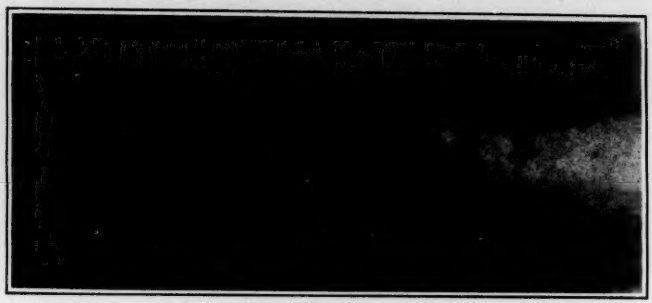

FIGURE 32.

FIGURE 31.

tivity of the method as applied to this particular job was very high.

The results of this visual inspection are shown in the next series of photographs. Figure 23 shows the section represented in film G (Figure 18) with a layer machined off of the surface. The heavy void noticeable in the upper part of this film is not visible in the sectioned piece because this area was welded after the radiographs were taken. The defects to the lower right of G_1, however, are those which may be seen in the vicinity of the letter G in film G.

The pieces shown sectioned in Figure 24 are those of the radiograph film H (Figure 19). Here again the major defect, the long crack, was welded, and in this figure part H_3 shows the built-up weld metal with cracks now appearing in the weld metal, and piece H_2 shows a crack beyond the limits of welding, therefore not repaired by the welding.

Figure 25 shows on piece I_2 the end of the crack shown in film I (Figure 20) which likewise had not been removed completely by the chipping and welding. These last few films illustrate the uncertainty in repairing cracks by visual examination and chipping. The material had been chipped out to remove the entire extent of the crack so far as the eye could tell, but the chipping process itself tended to smear the end of a fine crack. In many cases it has been found that the ends of the cracks had not been completely removed. It is obvious that sensible application of radiography to such cases would permit more intelligent repair of such defects as these.

Figure 26 shows the area represented in film C (Figure 21). Here may be discovered the circular crack so striking in film C with a fairly complete registration of the defects indicated on film C. The square marking on film C is not visible on this sectioned piece because it was a surface depression and not an interior defect. It should be remarked that visible inspection of sectioned pieces is not nearly so complete as radiographic inspection since an infinite number of sections would have to be prepared in order to get as complete a record of the internal state of the casting as that given by a radiograph.

Shortly after the application of the gamma ray method to the sternpost casting of the *Chester* an opportunity presented itself

to apply the method to a keel knuckle casting intended for the same class of cruisers.

This casting is shown in the next illustration, Figure 27, together with indications of the placement of the films. The thickness of the right hand section of this casting, along the right hand border of films A, B, and C, was between 2–1/2 and 3 inches. This thickness increased to 7 inches along the left hand border (as illustrated by the chalk marks) of films D, E, F, and H. After the manufacture of the casting a crack was discovered in the D, E, F, and H area, and welded. The welded area is shown by the bright area in the photograph. In this case again it was believed that the chipping and welding had removed the defective metal.

The application of the gamma ray method to this casting was of considerable interest since it presented opportunity to test the method on thicknesses far beyond the X-ray range and also upon sections of varying thickness.

The photographs shown in Figure 27 were all taken at one time. The radium, 350 millicuries, was placed behind the casting at a distance from the nearest film of about 15 inches and exposed overnight. Some of the radiographs obtained are shown in our next series of photographs.

Film A (Figure 28) shows the condition of the metal in section A of Figure 27. It may be seen that a general "cracky" condition existed in this area. Film D (Figure 29) represents the area D in Figure 27. The thickness at the right hand side of this film was 3 inches and at the left hand side 7 inches. The very heavy and sharp crack shown near the middle of the film is that under the bright portion which had been welded. This crack comes in the thickness of 5 inches, well beyond the X-ray range, and indicates that the welding had not been extensive enough to remove the defective metal. Film F (Figure 30), representing section F in the Figure 27, shows an extremely bad condition of the metal. Here again the thickness at the right-hand side is 3 inches and at the left hand side 7 inches. The large number of severe cracks shown in this film indicates the very poor condition of the metal. Indeed, films D, E, F, and H showed that the original crack had not been removed, for the radiographs regis-

tered a continuous series of cracks from the top of D to the bottom of H.

These films, then, show the efficiency of the testing method for sections beyond the X-ray limit. They indicate also the application of the method to objects of changing cross-section, for films D and F show a nearly correct exposure for both the right hand and left hand sides of the films, although the thickness changed from 3 inches to 7 inches over this distance. For a similar experiment with X-rays (in thickness ranges where the X-ray could have been used) it would have shown a very severe under-exposure at the thick section or a very severe over-exposure at the thin section, necessitating more than one exposure in order to obtain complete records. Because of the lack of sensitivity of the method to changing cross-section, mentioned above, this changed cross-section did not offer any particular difficulty.

It should not be assumed from the radiographs shown that all castings are as uniformly defective as the areas shown in the radiographs indicate. It is natural that in the demonstration of the performance of a non-destructive testing method only defective materials should be discussed. As a matter of fact, many radiographs have been taken, even upon the sternpost casting of the *Chester,* which showed no appreciable defects.

The correct application of such a testing method is that it should be for the determination of the location and nature of defects with a certain type of pattern design upon a pilot casting. With the record disclosed by the radiography, the design engineer, metallurgist, and the foundryman should collaborate to improve the design and pattern in such a way that the disclosed defects may be remedied. In certain cases where very large and expensive castings are used for important purposes, such as that of the sternpost casting, it seems advisable that the whole casting should be radiographed in each case, in order that there be no question of its performance in service. For such castings the cost of the inspection method is inconsiderable in comparison with the necessity that the casting perform satisfactorily. Of course, border line cases occasionally will be met in which defects are found but where the casting is shown to be not greatly defective. In such cases it would be necessary to decide whether or not the

material is too bad to use. No categorical opinion can be expressed on such cases. Such questions should be made the subject of a conference between experts on the behavior of material. Without a suitable radiographic inspection method, however, such experts could have no reasonable grounds upon which to discuss the question of the safety of the casting nor to decide upon the disposition of the casting.

Most of the work which has been done so far has been restricted to steel castings. Some work, however, has been done on the application of the method to welds and this application will be more extensive in the future. It might be interesting to the readers of this JOURNAL, however, to see what might be expected from the method in this regard. Figure 31 shows a gamma ray radiograph of a single-V butt weld in a 1 inch plate. It requires no great powers of divination to note that the weld is very porous and would be rejected by any conscientious engineer. Figure 32, however, shows a very satisfactory weld of the same type. Here only an extremely slight porosity can be noted and there is no question whatsoever that the weld should be adjudged as satisfactory.

The application of the method to welds has some disadvantages in comparison to the X-ray method and some advantages. For the mass inspection of a very large number of welds the rather long exposure time characteristic of the gamma ray method would hamper its application. However, in cases where the welds are rather difficult to reach, where it would be an extremely awkward if not impossible task to arrange an X-ray apparatus, the gamma ray method will have its application. It is obvious, of course, that on very heavy welded structures where the thickness runs beyond 3 inches the gamma ray method is the only method available.

CONCLUSION.

It is inevitable that this new method of radiographic inspection should be compared to the X-ray method. It has been the intention of the above account to present the features of the new method in some detail and at the same time to draw this comparison. This newer method offers advantages in greater simplicity and operation, almost complete portability, far greater penetration,

and more satisfactory performance on irregular sections. It suffers from the rather longer exposure times and from the lack of general availability of radium.

These longer exposure times can, however, be compensated at least partially by the opportunity for simultaneous radiography such as indicated in the radiography of the gunslide. The lack of a wide distribution of radium available for purposes of radiography cannot be immediately remedied. Indeed, the whole question of the economics of the method cannot at present be adequately discussed since there have been as yet no arrangements made for the industrial prosecution of the method. The cost of an inspection method is naturally of very great interest to industry and before it can be claimed this new method will have wide application, it will be necessary to show that its costs are not prohibitive. Such surveys as have already been made, however, indicate that a sensible prosecution of the method industrially should result in costs which are well within the admitted range of inspection costs. It will be some time before all the details are sufficiently well worked out that simple answers can be given to any of the questions of cost. It is of interest to note that the Navy has considered the matter carefully and has decided to provide the navy yards on the east coast radium for this purpose.

This radium will be held at the Naval Research Laboratory. The total amount will be one-half gram divided into three portions so that separate portions can be sent to the various yards as required, and indeed to any other place where it is desired to perform radiographic inspection. It will also be used at the Naval Research Laboratory in connection with research on steel castings when not required at the yards.

With the advantages which the gamma ray method of radiography seems to offer, it seems not unreasonable to expect that industry will find use for this new method and that in time it will take its place with other well established methods of non-destructive testing. In fact, the Research Corporation of New York are investigating its possibilities for commercial use and also the question of making the necessary facilities available.

ANALYSIS OF STRAIGHT LINE PLOTTING DATA.

By Lieutenant Leonard Kaplan (C. C.), U. S. N., Member.

There are three generally accepted methods of fairing a straight line through a set of plotted points. These are,

I The Graphic Method.
II The Method of Averages.
III The Method of Least Squares.

I THE GRAPHIC METHOD.

The matter of drawing the best representative straight line through a set of plotted points is perforce a matter of judgment, and is for that reason not susceptible of mathematical analysis. No discussion of its merits has been included in this paper, which is limited to an exposition of the so-called "calculated lines."

II THE METHOD OF AVERAGES.

(a) Line passing through the origin — Type equation $y = bx$

In the method of averages, the line is drawn so that the sum of the normal distances of the plotted points on the one side of the line, is equal to the sum of the normal distances of the plotted points on the other side of the line. It will be noted that these normal distances are proportional to the vertical distances, since the

$$\text{normal distance of any point } (x, y) \text{ to the line} = \frac{y - bx}{\sqrt{1 + b^2}}$$

where $\sqrt{1 + b^2}$ is the constant of proportionality.

By hypothesis, the method of averages line is one for which

$$\Sigma \frac{y - bx}{\sqrt{1 + b^2}} = 0$$

That is,

$$\Sigma y - b\Sigma x = 0$$

$$b = \frac{\Sigma y}{\Sigma x}$$

This is a symmetrical equation in x and y, for if we start by assuming

$$x = By, \quad (2)$$

that is, letting y be the independent variable instead of x, we have by analogy,

$$B = \frac{\Sigma x}{\Sigma y} \quad (1)'$$

and this substituted in our assumed relation gives,

$$x = \frac{\Sigma x}{\Sigma y} y,$$

which transposed is,

$$y = \frac{\Sigma y}{\Sigma x} x$$

or

$$y = bx,$$

our original type equation.

The method of averages line may be considered as an axis through the center of gravity of the points (assuming the points of equal weight) and therefore also an axis of zero moment.

The following example * is a fair illustration of the application of the method of averages for a line passing through the origin, type equation $y = bx$.

W	E
0	0
50	0.0130
100	0.0251
150	0.0387
200	0.0520
225	0.0589
250	0.0659
260	0.0689
Σ1235	Σ0.3225

By hypothesis,

$$E = bW$$

$$b = \frac{\Sigma E}{\Sigma W} = \frac{.3225}{1235} = .00026113 \dagger$$

and,

$$E = .00026113\, W$$

* This example has been taken from Joseph Lipka's "Graphical and Mechanical Computation," 1918, pp. 122-125, inclusive. W is the stretching load in pounds applied to a wire, E is the elongation ratio.

† It is realized that the value of b has been carried out to more decimal places than are warranted by the number of significant figures in the data. This has been done for purposes of comparison with other methods of plotting.

Consider next an example* of a line produced through the origin from the first quadrant into the third quadrant, where x and y have negative values.

tan θ	M
- .02241	- 9.321
- .02115	- 9.321
- .02027	- 9.321
.02106	9.521
.02250	9.521
.02208	9.521
- .03787	- 16.365
- .03803	- 16.365
- .03585	- 16.365
.03641	16.407
.03657	16.407
.03709	16.407
Σ .00013	Σ .726

By hypothesis,

$$\tan\theta = bM$$

$$b = \frac{\Sigma \tan\theta}{\Sigma M} = \frac{.00013}{.726} = .00017906$$

and,

$$\tan\theta = .00017906\ M$$

It is important to note that the summations of $\tan\theta$ and M in this case, are in effect differences of the summations of the plus and minus values of the coordinates of the plotted points. The number of significant figures in $\Sigma \tan\theta$ and ΣM has in each case been reduced by two, with consequent impairment of the accuracy of evaluation of the slope b.

(b) General case of a line that does not pass through the origin —Type equation $y = a + bx$

The application of the method of averages to this case necessitates the division of the data into two groups, from which two simultaneous equations in a and b are obtained and then solved. Obviously, there are a number of different ways in which the data may be divided into two groups. The values of a and b obtained by this method will depend therefore upon the division chosen. For this reason, no discussion of the application of the method of averages to the general case of a straight line, type equation $y = a + bx$ is presented herewith.†

* This example is taken from a recent (surface) inclining experiment of the U. S. S. *S-51*. M is the inclining moment in foot-tons (port = +) and Θ is the angle of inclination, also plus to port.

† Formulae for straight lines, type equation $y = a + bx$, by the method of averages, may be found in Joseph Lipka's "Graphical and Mechanical Computation," 1918, pp. 125-126, and also in "Engineering Mathematics" (published by the United States Naval Institute, 1926), Chapter VI, pp. 3-5.

III THE METHOD OF LEAST SQUARES.

(a) Line passing through the origin—Type equation $y = bx$

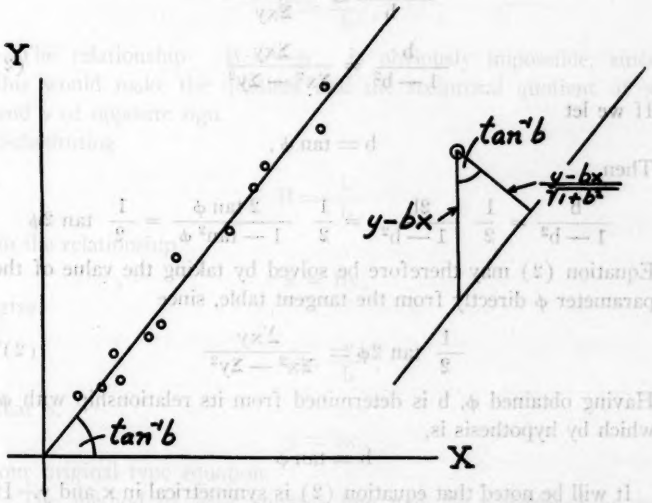

FIGURE 1

In the method of least squares, the line is drawn so that the sum of the squares of the normal distances of the plotted points from the line is a minimum. Translated into an equation, this gives us

$$\sum \left[\frac{y - bx}{\sqrt{1+b^2}}\right]^2 = \text{a minimum.}$$

Then,

$$\frac{d}{db} \sum \frac{(y-bx)^2}{1+b^2} = 0$$

$$\sum \frac{(b^2+1)\ 2\ (y-bx)\ (-x) - (y-bx)^2\ 2b}{(b^2+1)^2} = 0$$

$$\Sigma\ (b^2+1)\ (y-bx)\ (-x) - (y-bx)^2\ b = 0$$

$$-(b^2+1)\ \Sigma xy + b\ (b^2+1)\ \Sigma x^2 - b\Sigma y^2 + 2b^2\ \Sigma xy - b^3\ \Sigma x^2 = 0$$

$$(2b^2 - b^2 - 1)\,\Sigma xy + (b^3 + b - b^3)\,\Sigma x^2 - b\Sigma y^2 = 0$$

$$(b^2 - 1)\,\Sigma xy + b\,(\Sigma x^2 - \Sigma y^2) = 0$$

$$\frac{b^2 - 1}{b} = \frac{\Sigma y^2 - \Sigma x^2}{\Sigma xy}$$

$$\frac{b}{1 - b^2} = \frac{\Sigma xy}{\Sigma x^2 - \Sigma y^2} \qquad (2)$$

If we let

$$b = \tan \phi\,,$$

Then,

$$\frac{b}{1 - b^2} = \frac{1}{2}\,\frac{2b}{1 - b^2} = \frac{1}{2}\,\frac{2\tan\phi}{1 - \tan^2\phi} = \frac{1}{2}\tan 2\phi$$

Equation (2) may therefore be solved by taking the value of the parameter ϕ directly from the tangent table, since

$$\frac{1}{2}\tan 2\phi = \frac{\Sigma xy}{\Sigma x^2 - \Sigma y^2} \qquad (2)'$$

Having obtained ϕ, b is determined from its relationship with ϕ, which by hypothesis is,

$$b = \tan \phi$$

It will be noted that equation (2) is symmetrical in x and y. If we interchange the variables, that is, let y be the independent variable and x the dependent variable, the line as calculated from equation (2) with the variables interchanged, will coincide exactly with the one determined for x the independent variable and y the dependent variable. This can be demonstrated by letting

$$x = By$$

Then by analogy, we have

$$\frac{B}{1 - B^2} = \frac{\Sigma xy}{\Sigma y^2 - \Sigma x^2} \qquad (2)''$$

Combining equations (2) and (2)″ gives

$$\frac{b}{1 - b^2} = \frac{-B}{1 - B^2}$$

$$b - bB^2 = -B + b^2B$$

$$(b + B) - bB\,(b + B) = 0$$

$$(b + B)\,(1 - bB) = 0$$

and,

$$B = -b, \qquad \text{or}$$

$$B = \frac{1}{b}$$

The relationship $B = -b$ is obviously impossible, since this would make the quotient and the reciprocal quotient of x and y of opposite sign.

Substituting

$$B = \frac{1}{b}$$

in the relationship

$$x = By,$$

gives

$$x = \frac{y}{b},$$

that is,

$$y = bx,$$

our original type equation.

The method of least squares line may be considered as an axis of least moment of inertia, the individual points being regarded as of equal weight.

Applying this method to the first of the examples given above, we have

W	E	EW	W^2	E^2
0	0	0	0	0
50	0.0130	0.650	2,500	0.0001690
100	0.0251	2.510	10,000	0.0006300
150	0.0387	5.805	22,500	0.0014977
200	0.0520	10.400	40,000	0.0027040
225	0.0589	13.253	50,625	0.0034692
250	0.0659	16.475	62,500	0.0043426
260	0.0689	17.914	67,600	0.0047472
Σ1,235	0.3225	67.007	255,725	0.0175599

By hypothesis,

$$E = bW$$

$$\frac{b}{1-b^2} = \frac{\Sigma EW}{\Sigma W^2 - \Sigma E^2} = \frac{67.007}{255{,}725 - .0175599}$$

$$\frac{1}{2}\tan 2\phi = \frac{67.007}{255{,}725} = .00026203$$

Since ϕ is very small,

$$\frac{1}{2}\tan 2\phi = \frac{1}{2}(2\phi) = \phi = b \quad \text{(very nearly)}$$

That is,

$$b = .00026203$$

and

$$E = .00026203\ W$$

Computing the second of the examples given above, we have

$\tan\theta$	M	M $\tan\theta$	$\tan^2\theta$	M^2
- .02241	- 9.321	.20888361	.0005022081	86.881041
- .02115	- 9.321	.19713915	.0004473225	86.881041
- .02027	- 9.321	.18893667	.0004108729	86.881041
.02106	9.521	.20051226	.0004435236	90.649441
.02250	9.521	.21422250	.0005062500	90.649441
.02208	9.521	.21022368	.0004875264	90.649441
- .03787	- 16.365	.61974255	.0014341369	267.813225
- .03803	- 16.365	.62236095	.0014462809	267.813225
- .03585	- 16.365	.58668525	.0012852225	267.813225
.03641	16.407	.59737887	.0013256881	269.189649
.03657	16.407	.60000399	.0013373649	269.189649
.03709	16.407	.60853563	.0013756681	269.189649
Σ .00013	.726	4.85462511	.0110020649	2143.600068

By hypothesis,

$$\tan\theta = bM$$

$$\frac{b}{1-b^2} = \frac{\Sigma M \tan\theta}{\Sigma M^2 - \Sigma \tan^2\theta}$$

$$\frac{1}{2}\tan 2\phi = \frac{4.85462511}{2143.600068 - .0110020649} =$$

$$\frac{4.85462511}{2143.589066} = .0022647$$

$$b = \frac{1}{2} \tan 2\phi = .0022647$$

and,

$$\tan \theta = .0022647\ M$$

It is at once apparent that the evaluation of the constant of proportionality for tan θ and M by the method of least squares gives a result radically different from that obtained by the method of averages. The differences between the given and the computed values of tan θ for the specific values of M used are tabulated for the two methods in the following analysis.

It would seem to be a fair conclusion that the indiscriminate use of the method of averages in plotting straight lines may lead to erroneous results. In other words, a line drawn through the origin and the center of gravity of a set of plotted points (and this, it should be noted, is also what is generally attempted in "fairing" a line by eye) may be wide of the correct result.

It is true that in the first example given, the equations

(1) $E = .00026113\ W$ (by the Method of Averages)
(2) $E = .00026203\ W$ (by the Method of Least Squares)

check very closely. But it is a poor rule that applies only to special cases, and it would seem to be the better part of wisdom to adopt a formula which can be applied indiscriminately to all cases, and which does not require an investigation into its merits for use in a particular example.

For the general case of a straight line passing through the origin, use the formula

$$\frac{b}{1 - b^2} = \frac{\Sigma xy}{\Sigma x^2 - \Sigma y^2}, \qquad (2)$$

where

$$b = \tan \phi$$

and for facility of computation, may be obtained by use of trigonometric tables, from the relation

$$\frac{b}{1 - b^2} = \frac{1}{2} \tan 2\phi = \frac{\Sigma xy}{\Sigma x^2 - \Sigma y^2}$$

(b) General case of a line that does not pass through the origin —Type equation $y = a + bx$

		Method of Averages		Method of Least Squares			
		$(\tan\theta)_a = .00017906\ M$		$(\tan\theta)_s = .0022647\ M$			
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
$\tan\theta$	M	$(\tan\theta)_a$	$\Delta_a=(1)-(3)$	$(\tan\theta)_s$	$\Delta_s=(1)-(4)$	Δ_a^2	Δ_s^2
- .02241	- 9.321	- .00167	- .02074	- .02111	- .00130	.0004301476	.0000016900
- .02115	- 9.321		- .01948		- .00004	.0003794704	.0000000016
- .02027	- 9.321		- .01860		.00084	.0003459600	.0000007056
.02106	9.521	.00170	.01936	.02156	- .00050	.0003746096	.0000002500
.02250	9.521		.02080		.00094	.0004326400	.0000008836
.02206	9.521		.02038		.00052	.0004153444	.0000002704
- .03787	- 16.365	- .00293	- .03494	- .03706	- .00081	.0012208036	.0000006561
- .03603	- 16.365		- .03510		- .00097	.0012320100	.0000009409
- .03585	- 16.365		- .03292		.00121	.0010837264	.0000014641
.03641	16.407	.00294	.03347	.03716	- .00075	.0011191409	.0000005625
.03657	16.407		.03363		- .00059	.0011309769	.0000003481
.03709	16.407		.03415		- .00007	.0011662225	.0000000049
Σ.00013	.726		.00001		- .00152	.0000077778	.0093312523
		Σ_{Δ_a} (+ & -) .32357		Σ_{Δ_s} (+ & -) .00854			
		Average Δ_a .02696		Average Δ_s .00071			
		$\Sigma\,\Delta_a^2$ .00933		$\Sigma\,\Delta_s^2$ .00000778			

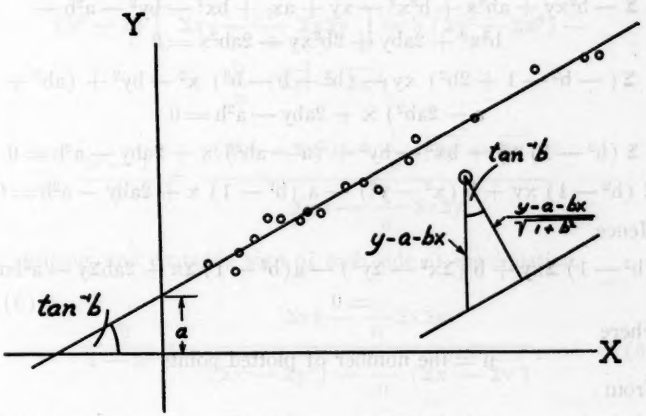

FIGURE 2

As in the case of a line passing through the origin, the line is drawn so that the sum of the squares of the normal distances of the plotted points from the line is a minimum. This gives us the equation

$$\sum\left[\frac{y-a-bx}{\sqrt{1+b^2}}\right]^2 = \text{a minimum}$$

Then,

$$\sum\frac{(y-a-bx)^2}{1+b^2} = \text{a minimum}$$

and,

$$\frac{d}{db}\sum\frac{(y-a-bx)^2}{1+b^2} = 0$$

$$\sum\frac{(1+b^2)\ 2\ (y-a-bx)\ (-x) - (y-a-bx)^2\ (2b)}{(1+b^2)^2} = 0$$

$$\Sigma\ (1+b^2)\ (y-a-bx)\ (-x) - (y-a-bx)^2\ b = 0$$

$$\Sigma\ (b^2y - ab^2 - b^3x + y - a - bx)\ (-x) - (y^2 + a^2 + b^2x^2 - 2ay - 2bxy + 2abx)\ b = 0$$

$$\Sigma - b^2xy + ab^2x + b^3x^2 - xy + ax + bx^2 - by^2 - a^2b - b^3x^2 + 2aby + 2b^2xy - 2ab^2x = 0$$

$$\Sigma\,(-b^2 - 1 + 2b^2)\,xy - (b^3 + b - b^3)\,x^2 - by^2 + (ab^2 + a - 2ab^2)\,x + 2aby - a^2b = 0$$

$$\Sigma\,(b^2 - 1)\,xy + bx^2 - by^2 + (a - ab^2)\,x + 2aby - a^2b = 0$$

$$\Sigma\,(b^2 - 1)\,xy + b\,(x^2 - y^2) - a\,(b^2 - 1)\,x + 2aby - a^2b = 0$$

Hence,

$$(b^2 - 1)\,\Sigma xy + b\,(\Sigma x^2 - \Sigma y^2) - a(b^2 - 1)\,\Sigma x + 2ab\Sigma y - a^2bn = 0 \qquad (3)$$

where

$$n = \text{the number of plotted points}$$

From

$$\frac{d}{da}\sum\frac{(y - a - bx)^2}{1 + b^2} = 0,$$

we have

$$\Sigma - 2\,(y - a - bx) = 0,$$

and

$$a = \frac{\Sigma y - b\Sigma x}{n} \qquad (4)$$

Substituting the value of a from (4) in (3),

$$(b^2 - 1)\,\Sigma xy + b\,(\Sigma x^2 - \Sigma y^2) - \frac{b^2 - 1}{n}\left[\Sigma x\Sigma y - b\overline{\Sigma x}^2\right] + \frac{2b}{n}\left[\overline{\Sigma y}^2 - b\Sigma x\Sigma y\right] - \frac{b}{n}\left[\overline{\Sigma y}^2 - 2b\,\Sigma x\Sigma y + b^2\,\overline{\Sigma x}^2\right] = 0$$

Combining,

$$(b^2 - 1)\,\Sigma xy + b\,(\Sigma x^2 - \Sigma y^2) - \left[\frac{b^2 - 1}{n}\right]\Sigma x\Sigma y + \frac{1}{n}\,(b^3 - b - b^3)\,\overline{\Sigma x}^2 + \frac{1}{n}(2b - b)\,\overline{\Sigma y}^2 = 0$$

$$(b^2 - 1)\,\Sigma xy + b\,(\Sigma x^2 - \Sigma y^2) - \left[\frac{b^2 - 1}{n}\right]\Sigma x\Sigma y + \frac{b}{n}(\overline{\Sigma y}^2 - \overline{\Sigma x}^2) = 0$$

$$(b^2-1)\left[\Sigma xy-\frac{1}{n}\Sigma x\Sigma y\right]=b\,(\Sigma y^2-\Sigma x^2)-\frac{b}{n}\,(\overline{\Sigma y}^2-\overline{\Sigma x}^2)$$

$$\frac{b^2-1}{b}=\frac{(\Sigma y^2-\Sigma x^2)-\frac{1}{n}\,(\overline{\Sigma y}^2-\overline{\Sigma x}^2)}{\Sigma xy-\frac{1}{n}\Sigma x\Sigma y}$$

Capsizing, and changing sign of each side of the equation,

$$\frac{b}{1-b^2}=\frac{\Sigma xy-\frac{1}{n}\Sigma x\Sigma y}{(\Sigma x^2-\Sigma y^2)-\frac{1}{n}\,(\overline{\Sigma x}^2-\overline{\Sigma y}^2)} \qquad (5)$$

Here, too, b can be evaluated more conveniently by means of the trigonometric substitution

$$\frac{b}{1-b^2}=\frac{1}{2}\tan 2\phi\,,$$

where

$$b=\tan\phi$$

The value of the constant a may be determined then from equation (4), b now being known, since

$$a=\frac{\Sigma y-b\Sigma x}{n} \qquad (4)$$

It will be noted that equation (5) like equation (2) is symmetrical in x and y. If we interchange the variables, that is, let y be the independent variable and x the dependent variable, the line as calculated from equation (5) with the interchanged variables will coincide exactly with the one determined from this equation with x the independent variable and y the dependent variable.

The proof is similar to that given for equation (2). The effect of interchanging the variables is to change the sign of the right hand side of the equation, making

$$\frac{b}{1-b^2}=\frac{-B}{1-B^2}\,,$$

where B is the slope of the line referred to the y axis instead of the x axis. As before, it can be shown that

$$B = \frac{1}{b},$$

which means that the two lines have the same directional slope. As defined above,

$$a = \frac{\Sigma y - b\Sigma x}{n}, \qquad (4)$$

where a is the y intercept of the line $y = a + bx$. Interchanging the variables, we have

$$x = A + By$$

By analogy with equation (4),

$$A = \frac{\Sigma x - B\Sigma y}{n} \qquad (4)'$$

Then,

$$\frac{A}{a} = \frac{\Sigma x - B\Sigma y}{\Sigma y - b\Sigma x}$$

and since

$$B = \frac{1}{b},$$

$$\frac{A}{a} = -\frac{1}{b}\,\frac{(\Sigma y - b\Sigma x)}{(\Sigma y - b\Sigma x)} = -\frac{1}{b}$$

Substituting their equivalents for A and B in equation (4)', we have

$$x = -\frac{a}{b} + \frac{y}{b},$$

which, when cleared of fractions, gives

$$y = ax + b$$

our type equation, proving that the relation of x and y as obtained by interchanging the variables is identical with the original one where x was the independent variable.

The following example * illustrates the application of the method of least squares for the general case of a line which does not pass through the origin, type equation $y = ax + b$.

C	r	C^2	r^2	rC
19.1	76.30	364.81	5821.6900	1457.330
25.0	77.80	625.00	6052.8400	1945.000
30.1	79.75	906.01	6360.0625	2400.475
36.0	80.80	1296.00	6528.6400	2908.800
40.0	82.35	1600.00	6781.5225	3294.000
45.1	83.90	2034.01	7039.2100	3783.890
50.0	85.10	2500.00	7242.0100	4255.000
245.3	566.00	9325.83	45825.9750	20044.495

By hypothesis,

$$r = a + bC$$

$$\frac{b}{1 - b^2} = \frac{\Sigma rC - \frac{1}{n}\Sigma C \Sigma r}{(\Sigma C^2 - \Sigma r^2) - \frac{1}{n}(\overline{\Sigma C}^2 - \overline{\Sigma r}^2)}$$

$$= \frac{20{,}044.495 - \frac{1}{7}(245.3)(566.00)}{(9325.83 - 45{,}825.975) - \frac{1}{7}\left[(245.3)^2 - (566.00)^2\right]}$$

$$= \frac{140{,}311.465 - 138{,}839.800}{7\ (-36{,}500.145) - (60{,}172.09 - 320{,}356.00)}$$

$$= \frac{1471.665}{-255{,}501.015 + 260{,}183.91} = \frac{1471.665}{4682.895}$$

$$\tan 2\phi = \frac{2b}{1 - b^2} = (2)\ (.31427) = .62853$$

$$2\phi = 32° - 9'.036$$

$$\phi = 16° - 4'.518$$

$$b = \tan \phi = .28817$$

* This example has been taken from Joseph Lipka's "Graphical and Mechanical Computation," 1918, pp. 125-127, inclusive. r is the resistance in ohms of a wire, C the temperature rise in degrees Centigrade.

$$a = \frac{\Sigma r - b\Sigma C}{n} = \frac{566.00 - .28817\ (245.3)}{7}$$

$$= \frac{566.00 - 70.6881}{7} = \frac{495.3119}{7} = 70.759$$

Hence,

$$r = 70.759 + .28817\ C$$

For the second example, consider again the inclining experiment problem, M and $\tan\theta$, which has already been computed first by the method of averages, and second by the method of least squares, as a line passing through the origin. Theoretically, *i.e.*, from the mechanics of the inclining experiment, this line should pass through the origin. Practically, the amount by which it fails to pass through the origin is generally accepted as a measure of the accuracy of the experiment. To find this distance, let us determine the equation of the line (by the method of least squares) passing through the points, type equation $y = a + bx$.

By hypothesis,

$$\tan\theta = a + bM$$

From equation (5), we have

$$\frac{b}{1 - b^2} = \frac{\Sigma M \tan\theta - \frac{1}{n}\Sigma M \Sigma \tan\theta}{(\Sigma M^2 - \Sigma \tan^2\theta) - \frac{1}{n}(\Sigma M^2 - \Sigma \tan^2\theta)}$$

$$= \frac{4.85462511 - \frac{1}{12}(.726)\ (.00013)}{(2143.600068 - .0110020649) - \frac{1}{12}\left[(.726)^2 - (.00013)^2\right]}$$

$$= \frac{58.25550132 - .00009438}{(12)\ (2143.5890659351) - (.527076 - .0000000169)}$$

$$= \frac{58.25540694}{25723.0687912212 - .5270759831} = \frac{58.25540694}{25722.5417152381}$$

$$= .0022648$$

and since b is small,

$$b = \frac{b}{1 - b^2} = .0022648 \quad \text{(very nearly)}$$

From equation (4), we have

$$a = \frac{\Sigma \tan\theta - b\Sigma M}{n}$$

$$= \frac{.00013 - .0022648\ (.726)}{12}$$

$$= \frac{.00013 - .0016443}{12} = \frac{-.0015143}{12}$$

$$= -.00012619$$

From which we have

$$\tan\theta = -.00012619 + .0022648\ M \quad \text{(Method of Least Squares)}$$

This compares with the relation

$$\tan\theta = .0022647\ M \quad \text{(Method of Least Squares)}$$

deduced above for a line passing through the origin. It is interesting to note that the values of b as determined in both cases are sensibly the same.

DISCUSSION AND CONCLUSION.

It is the opinion of the author of this paper that lines worth calculating should be calculated by the Method of Least Squares. The formulae deduced by this method, equation (2) for a line passing through the origin, type equation $y = bx$, and equations (4) and (5) for the general case, type equation $y = a + bx$, may be used indiscriminately for calculating any linear relationship.

The point has been stressed that the formulae presented herein give symmetrical lines; that is, lines calculated by assuming either variable the independent variable will lie in the same relative position to the plotted points, which is another way of saying that the equations of the lines as determined by the two assumptions will be identical.

It is in respect to symmetry that the equations developed in this paper are different from those ordinarily presented in most textbooks on the subject. These generally concern themselves with the so-called "residuals," the differences that is, of the observed or given values of the dependent variable y from the calcu-

lated ones. The effect of this is to establish a line through the plotted points such that the sum of the squares of the vertical distances of the points from the line is a minimum. Following this method, equation (2) for a line passing through the origin becomes

$$b = \frac{\Sigma xy}{\Sigma x^2} \qquad (2A)$$

and equation (5) for the general case of a line which does not pass through the origin, becomes

$$b = \frac{\Sigma x \Sigma y - n\Sigma xy}{\Sigma x^2 - n\Sigma x^2} \qquad (5A)$$

the relationship for a, equation (4), remaining the same, namely,

$$a = \frac{\Sigma y - b\Sigma x}{n} \qquad (4)$$

Taking the examples given above and computing them by the non-symmetrical formulae (2A) and (5A), we have the following comparisons:

(a)	Symmetrical	(Eq. 2)	$E = .00026203$ W
	Non-symmetrical	(Eq. 2A)	$E = .00026203$ W
(b)	Symmetrical	(Eq. 2)	$\tan\theta = .0022647$ M
	Non-symmetrical	(Eq. 2A)	$\tan\theta = .0022647$ M
(c)	Symmetrical	(Eq. 5)	$r = 70.759 + .28817$ C
	Non-symmetrical	(Eq. 5A & 4)	$r = 70.762 + .28807$ C
(d)	Symmetrical	(Eq. 5)	$\tan\theta = -.00012619 + .0022648$ M
	Non-symmetrical	(Eq. 5A & 4)	$\tan\theta = -.00012620 + .0022650$ M

It is evident from the above that the non-symmetrical formulae for the examples cited, give results almost the same as those of the symmetrical formulae. However, the plotting data in each of the examples selected is relatively good, and large discrepancies in the two methods of analysis might not therefore logically be expected. The additional work involved in calculating the sym-

metrical equation is relatively small, consisting principally of one extra column summation. The calculation of b from

$$\frac{b}{1-b^2}$$

is relatively simple when the tangent table is used; and where b is small,

$$b = \frac{b}{1-b^2} \text{ very nearly.}$$

It is probably a fair statement to say that the formula is best which requires the least thinking for its application. Even though it may involve additional computation, it requires less real mental effort and offers less opportunity for error, since computation is a mechanical function which can be checked in routine manner. For these reasons, and because they are more truly applicable to the sense or logic of the problem, the author recommends the use of the symmetrical "least squares" equations in all cases where straight lines are calculated from plotting data.

THE TREND OF NAVAL ENGINEERING, NO. 4, SOUND VERSUS SILENT MOTION PICTURES.

By C. Huey, Civil Member.*

As is generally known, the recent trend of the commercial motion picture industry has been from the silent motion pictures, to the sound or "talking" motion pictures. From the standpoint of the U. S. Navy the transition has taken place during the year 1931. In view of the widespread interest shown by the personnel of the Navy, and the associated public, it is of interest to describe and illustrate the fundamental features involved in both systems and to explain the engineering and economic reasons which resulted in the technical specifications covering the present form of Sound Motion Picture equipment.

The silent motion pictures have been in existence in the Navy since 1915. These equipments were, at first, purchased by the personnel of the Navy from community funds. In 1923 the purchase and upkeep was taken over by one of the Navy technical Bureaus under whose direction extensive tests were conducted on the best known commercial makes, to obtain data for detailed specifications. From the result of purchases under technical specifications, the standard of performance and quality was raised, the costs in general were reduced, and the service life of the films was greatly extended.

In the year 1929 the sound motion pictures had been conclusively established for commercial entertainment. The outlook for obtaining future supplies of new pictures for the Navy rapidly diminished as the Sound Motion Pictures supplanted the Silent Motion Pictures in the show houses. The reaction of the Navy personnel, as to be expected, was unanimously for the sound or "talking" motion pictures with its wealth of added entertainment.

Shortly after the Navy was definitely committed to Sound Motion Pictures, investigation was made to determine the mechanism required to produce the sound feature and to determine to what

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extent the silent motion picture equipments in the service could be incorporated or modified to meet the requirements for the acceptable performance of sound.

Investigation of the most popular makes of "sound on film" motion picture equipment revealed that most of the mechanism of standard silent motion pictures machines was being used, to which was added the necessary mechanism for the sound features. The fundamental difference between the film travel of the two systems is shown in simplified form in Figure 1. In both systems, the passage of the film from the supply reel to the take-up reel is identical as regards the picture projection, the optical system, the motor-driven intermittent movement, fire protection, and the illuminating unit. On the sound system, however, there is added an additional film loop and direction sprockets, an exciter lamp for the sound track, an additional optical system with aperture for the sound track, a photo-electric cell which converts variable light into electrical impulses, and a mask frame to blank off the sound track of the film from the picture projection system. In addition, a precision constant speed motor with variable speed control is substituted for the usual driving motor in order that the proper sound frequency standard may be maintained.

From the foregoing, it would appear that for economical reasons, the silent motion picture equipment in the service could readily be converted into sound motion picture machines. Detailed tests of complete sound motion picture equipment revealed that it would be impractical to enforce, by contract, the satisfactory performance of a converted equipment in which was incorporated such parts as were in common with the Navy's silent motion picture machines. The equipments in Naval service, which consisted of four or more distinct types or brands, were, in general, in a varying state of deterioration from age and usage, and would require complete overhaul before being applicable to conversion. The difficulty of fixing responsibility for the performance after overhaul, the necessity of re-calling all machines from the service, the long period of time involved during which the service would be without any picture entertainment, and the relatively small refund allowed toward the purchase price of complete sound motion picture equipment, resulted in the decision to purchase complete *new* equipment. The responsibility for the entire opera-

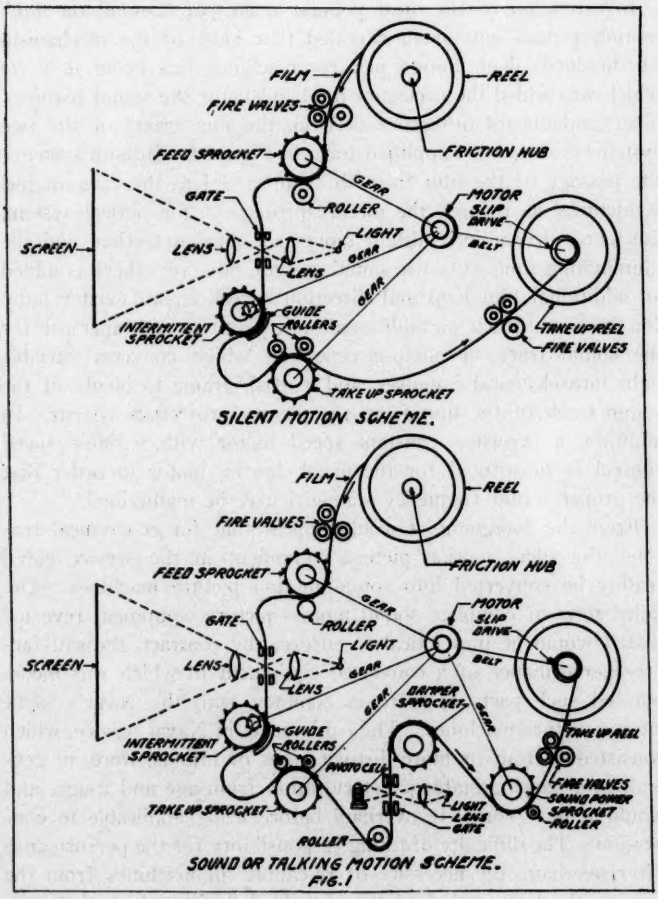

FILM
REEL
FIRE VALVES
FRICTION HUB
FEED SPROCKET
GEAR
GATE
ROLLER
MOTOR SLIP DRIVE
SCREEN
LENS
LIGHT
BELT
LENS
GEAR
GEAR
GUIDE ROLLERS
INTERMITTENT SPROCKET
TAKE UP REEL
FIRE VALVES
TAKE UP SPROCKET
SILENT MOTION SCHEME.
FILM
REEL
FIRE VALVES
FRICTION HUB
FEED SPROCKET
GEAR
MOTOR SLIP DRIVE
GATE
ROLLER
LIGHT
SCREEN
LENS
BELT
LENS
GEAR
GEAR
BAFFLER SPROCKET
GEAR
TAKE UP REEL
GUIDE ROLLERS
INTERMITTENT SPROCKET
PHOTO CELL
FIRE VALVES
SOUND POWER SPROCKET
ROLLER
LIGHT
LENS
GATE
TAKE UP SPROCKET
ROLLER
SOUND OR TALKING MOTION SCHEME.

FIG. 1

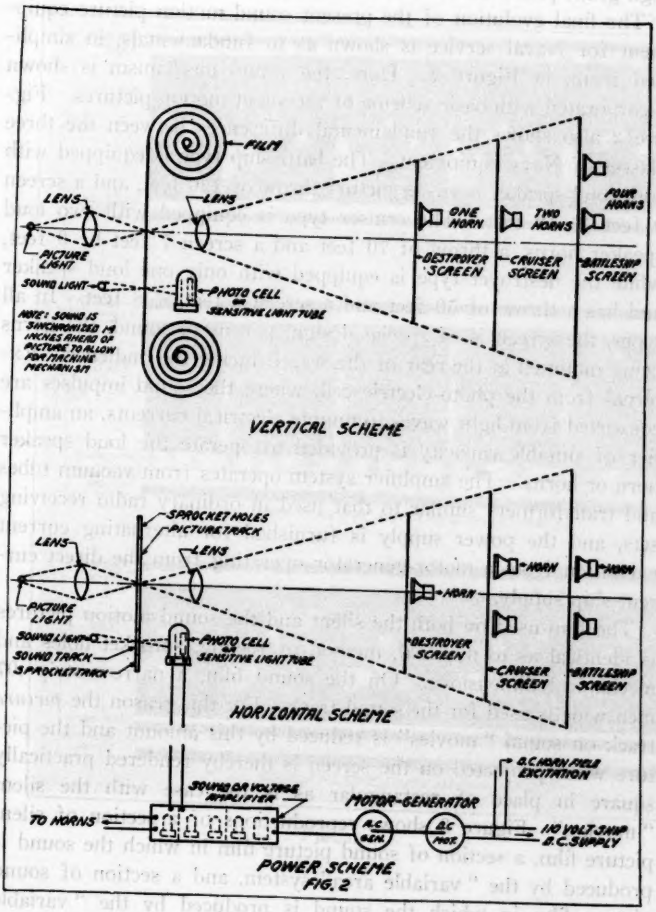

FILM
LENS
LENS
PICTURE LIGHT
SOUND LIGHT
PHOTO CELL
SENSITIVE LIGHT TUBE
NOTE: SOUND IS SYNCHRONIZED 14 INCHES AHEAD OF PICTURE TO ALLOW FOR TRACKING MECHANISM
ONE HORN
TWO HORNS
FOUR HORNS
DESTROYER SCREEN
CRUISER SCREEN
BATTLESHIP SCREEN
VERTICAL SCHEME
SPROCKET HOLES
PICTURE TRACK
LENS
LENS
PICTURE LIGHT
SOUND LIGHT
SOUND TRACK
SPROCKET TRACK
PHOTO CELL
SENSITIVE LIGHT TUBE
HORN
HORN
HORN
HORN
DESTROYER SCREEN
CRUISER SCREEN
BATTLESHIP SCREEN
HORIZONTAL SCHEME
D.C. HORN FIELD EXCITATION
SOUND OR VOLTAGE AMPLIFIER
MOTOR-GENERATOR
TO HORNS
A.C. GEN.
D.C. MOT.
110 VOLT SHIP D.C. SUPPLY
POWER SCHEME

FIG. 2

tion was placed on a contract basis which in turn was written around a rigid technical specification assuring suitable materials, high grade performance and reasonable life.

The final evolution of the present sound motion picture equipment for Naval service is shown as to fundamentals, in simplified form, in Figure 2. Here, the sound mechanism is shown incorporated with basic scheme of the silent motion pictures. Figure 2 also shows the fundamental differences between the three classes of Navy equipment. The battleship type is equipped with four loud speaker horns, a picture throw of 120 feet, and a screen 9 feet × 12 feet. The cruiser type is equipped with two loud speaker horns, a throw of 70 feet and a screen 7 feet by 9 feet, while the destroyer type is equipped with only one loud speaker and has a throw of 50 feet and a screen 6 feet × 8 feet. In all types, the screen is of special design, porous to sound, the horns being mounted at the rear of the screen facing the audience. External from the photo-electric cell, where the sound impulses are converted from light waves to minute electrical currents, an amplifier of suitable capacity is provided to operate the loud speaker horn or horns. The amplifier system operates from vacuum tubes and transformers similar to that used in ordinary radio receiving sets, and the power supply is furnished for alternating current excitation from a motor-generator operating from the direct current ship supply.

The film used on both the silent and the sound motion pictures is identical as to material, method of reeling, sprocket holes and mechanical dimensions. On the sound film, a narrow strip .07 inch wide is used for the sound track. For this reason the *picture* track on sound "movies" is reduced by this amount and the picture when projected on the screen is thereby rendered practically square in place of rectangular as is the case with the silent "movies." Figure 3 shows reproductions of a section of silent picture film, a section of sound picture film in which the sound is produced by the "variable area" system, and a section of sound picture film in which the sound is produced by the "variable density" system. The films of both of these sound systems are interchangeable in the Navy equipment, and the operation is unaffected if change is made from one system to the other as required, depending on which system is employed by the film producer.

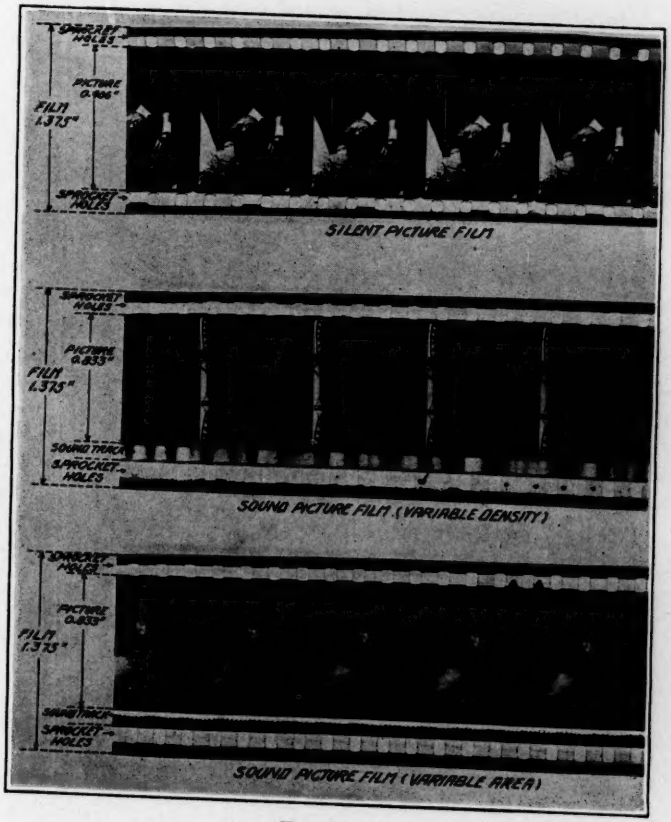

SPROCKET HOLES
PICTURE 0.906"
FILM 1.375"
SPROCKET HOLES
SILENT PICTURE FILM
SPROCKET HOLES
PICTURE 0.833"
FILM 1.375"
SOUND TRACK
SPROCKET HOLES
SOUND PICTURE FILM (VARIABLE DENSITY)
SPROCKET HOLES
PICTURE 0.833"
FILM 1.375"
SOUND TRACK
SPROCKET HOLES
SOUND PICTURE FILM (VARIABLE AREA)

FIGURE 3.

PROGRESS IN AIRSHIP DESIGN FROM U. S. S. *SHENANDOAH* TO U. S. S. *AKRON*.

By C. P. Burgess.*

The U. S. S. *Akron* is the second rigid airship to be constructed in the United States, unless that honor is granted to the small metalclad airship *ZMC-2* which in most respects resembles the nonrigid rather than the rigid type. The first American rigid airship was the U. S. S. *Shenandoah,* completed in 1923. A year later, the *Los Angeles,* a similar type of ship, built in Germany, was commissioned as a nonmilitary airship in the U. S. Navy.

From 1923 to 1931 the progress has been immense, in spite of the scarcity of intermediate steps of development. The progress may be most conveniently shown by a table of the principal characteristics of the three airships, U. S. S. *Shenandoah, Los Angeles,* and *Akron.* Table 1 shows the comparative strength and performance of these airships. The significance of the strength factors is explained later in the text.

It is interesting and instructive to analyze the changes from which such great improvements in performance have resulted; and to discover to what extent the progress is due to mere increase in size.

COMPARATIVE WEIGHTS.

The first step in the analysis is a comparison of the various items of weight, considered as absolute weights, and as fractions of the gross lift. This comparison is made in Table 2 for the complete ships; and in Table 3 for the largest main bays. To the designer, Table 3 showing the detailed structural weights in a single bay is the more interesting.

It may be surprising to find that the percentage of lift allotted to the longitudinals, which are the chief elements of strength against aerodynamic forces, has progressively diminished; while

* Principal Engineer, Bureau of Aeronautics.

TABLE 1

SOME COMPARATIVE FACTORS OF STRENGTH AND PERFORMANCE IN U.S.S. SHENANDOAH, LOS ANGELES AND AKRON.

	SHENANDOAH	LOS ANGELES	AKRON
Gross lift, lbs.	124,500	153,000	403,000
Useful lift, lbs.	44,100	62,600	170,000
Useful/gross	.354	.410	.422
Fuel load, lbs.	26,000	40,000	127,000
Fuel per hour at 50kts., lbs.	580	510	705
Endurance at 50 kts., hrs.	45	78.5	180
Propulsive coeff. at 50 kts.	40	44	60
Specific fuel consumption, lbs./ hp/hr.	.55	.47	.47
Horse-power at 50 kts.	1,050	1,090	1,500
Air volume, cu. ft.	2,300,000	2,800,000	7,250,000
Designed speed, kts.	50	68	72.8
" " ft./sec.	85	115	123
Max.velocity of sudden gust, ft./sec.	17.0	20.6	56.0
Bending moment due to gust, ft./lbs.	650,000	1,300,000	9,750,000
Max. stress due to gust, lbs./sq.in.	3,800	5,000	10,800
Static and secondary stresses, lbs./sq.in.	9,700	9,900	5,300
Max. total stress, lbs./sq.in.	13,500	14,900	16,100
Ult.strength of longitudinals, lbs./sq.in.	26,000	33,000	37,000
Factor of safety, lbs./sq.in.	1.9	2.2	2.3

TABLE 2

COMPARATIVE WEIGHTS OF U.S.S.SHENANDOAH, LOS ANGELES AND AKRON.

	SHENANDOAH		LOS ANGELES		AKRON	
	Weight lbs.	% Lift	Weight lbs.	% Lift	Weight Lbs.	% Lift
Hull structure	30,363	24.39	31,310	20.46	91,622	22.73
Fins, rudders, etc.	4,086	3.28	3,190	2.08	12,680	3.15
Cover, gas cells, valves, etc.	16,039	12.88	17,650	11.54	36,688	9.10
Power plant	14,849	11.93	20,800	13.59	50,412	12.51
Water recovery	2,055	1.65	2,332	1.52	10,400	2.58
Fuel system	3,204	2.57	2,610	1.71	7,100	1.76
Ballast system	906	0.73	627	0.41	2,950	0.73
Electrical, radio, etc.	1,119	0.90	2,555	1.67	3,760	0.93
Quarters and compartments	672	0.54	2,270	1.48	9,687	2.41
Mooring and handling	2,130	1.71	1,410	0.92	3,150	0.78
Control car and miscellaneous	4,980	4.00	5,675	3.71	4,451	1.10
Total weight empty	80,403	64.58	90,429	59.10	232,900	57.79
Gross lift at 95% full of gas lifting .062 lb./cu.ft.	124,500		153,000		403,000	

TABLE 3

STRUCTURAL WEIGHTS OF ONE MAIN FRAME BAY OF U.S.S.SHENANDOAH, LOS ANGELES AND AKRON.

	SHENANDOAH		LOS ANGELES		AKRON	
	Weight	% Lift	Weight	% Lift	Weight	% Lift
Longitudinals	635	7.24	940	5.40	2920	5.12
Main frame	431	4.91	748	4.30	3350	5.88
Intermediate frames	218	2.49	458	2.63	1650	2.89
Keel frames	58	0.66	154	0.89	1440	2.53
Total framing	1342	15.30	2300	13.22	9360	16.42
Transverse wires	112	1.28	156	0.90	--	--
Shear wires	72	0.82	174	1.00	525	0.92
Gas cell wires	90	1.03	74	0.43	560	0.98
Total wiring	274	3.13	404	2.33	1085	1.90
Gas cells	700	7.98	885	5.09	2500	4.39
Gas valves	35	0.40	38	0.22	300	0.53
Cordage netting	24	0.27	56	0.32	70	0.12
Exhaust trunk	23	0.26	24	0.14	--	--
Outer cover	260	2.96	633	3.64	1675	2.94
Total miscellaneous	1042	11.87	1636	9.40	4545	7.98
Grand total	2658	30.30	4340	24.96	14,990	26.30
Gross lift at 95% full of gas lifting .062 lb./cu.ft.	8770		17,400		57,000	

in other respects the *Los Angeles* has relatively the lightest framing, and the *Akron* the heaviest of the three ships. The reason for the diminishing weight of the longitudinals, in spite of greatly increased strength against aerodynamic forces may be seen from Table 1. The longitudinals of the *Akron* are able to sustain a stress of 10,800 pounds per square inch with a higher overall factor of safety than the *Shenandoah,* although the stress due to aerodynamic bending in the older airship was only 3,800 pounds per square inch. This great difference is the result of having girders more than 50 per cent stronger in proportion to their cross-sectional areas, and an improved wiring system which greatly diminishes the secondary bending stresses due to gas pressure.

The great weight of the main frames of the *Akron* is the result of their being of a novel type, consisting of a trussed ring about 8 feet deep and 8 feet wide, without transverse wire bracing, except a light resilient bulkhead to restrain the longitudinal surging of the gas cells. This type of frame gives a considerable degree of accessibility to the hull structure and gas cells while in flight; and it eliminates the danger of weakening the frame by breakages of transverse wires, which have proved somewhat troublesome in the *Los Angeles.* Each ring frame reduces the possible gas space by about 14,000 cubic feet, which would have a lift of 820 pounds if 95 per cent full of gas lifting, .062 pound per cubic foot. A still further weight penalty would have been paid for these frames if their spacing had not been increased so that the largest gas cell of the *Akron* has 14.1 per cent of the total lift of the ship, as compared with 11.4 per cent in the *Los Angeles,* and 7.0 per cent in the *Shenandoah.* From these figures it may be seen that the degree of sub-division has progressively diminished; and while this is in some respects a retrograde step from the standpoint of safety, it is not so bad as it looks from the figures, because in the later ships, there is a progressively increasing percentage of lift available for ballast which may be dropped in an emergency, and the greater engine power gives relatively more dynamic lift to overcome an excess of weight over buoyancy. Moreover, the progressive reduction in the ratio of length to diameter makes the later ships less sensitive to changes of trim by loss of gas near the ends.

When comparing the weights of the main transverse frames in Table 3, the weights of the transverse wiring in the *Shenandoah* and *Los Angeles* should be taken into account. The *Akron* has no structural wiring in the frames; and the weight of the resilient bulkhead separating adjacent gas cells is included in the main frame.

The percentage of gross lift absorbed by the intermediate frames shows a small but uniformly progressive increase; as might be expected from their increasing number, one per main bay in the *Shenandoah*, two in the *Los Angeles*, and three in the *Akron*.

The relative weights of the gas cell wiring and shear wires in the *Shenandoah* and *Akron* are almost the same; but the wires are much more efficiently disposed in the *Akron* so as to contribute to the longitudinal strength, and to reduce secondary stresses.

The rapidly increasing size of the gas cells in passing from the *Shenandoah* to the *Akron* has resulted in a considerable saving of weight in this item, obtained to some extent at the cost of reduced sub-division.

COMPARATIVE STRENGTH.

Table 1 shows the maximum velocities of sudden gusts which the three airships can encounter with the factors of safety listed in the last horizontal row of that table. It is assumed that when the ship is proceeding at the designed speed given in the table, a sharply defined gust is encountered in any direction in the plane at right angles to the initial direction of the ship's longitudinal axis. The maximum bending moment due to this condition is given approximately by the expression:

$$M = .09\rho uvV$$

Where M = the bending moment in feet pounds
ρ = the density of the air in pounds seconds ^{2}/feet ^{4}
u = the velocity of the gust in feet/seconds
v = the speed of the ship in feet/seconds
V = the air volume of the ship in cubic feet.

Only the *Akron* was designed on the basis of a gust of this nature. The gust velocities for the other two ships are determined by working backward from their maximum designed aerodynamic

bending moments, which were based on considerations of rudder and elevator forces, and dynamic lift. The *Akron* can withstand a sudden gust velocity 3.3 times greater than the limiting gust for the *Shenandoah;* but as already explained, improved design enables this greater strength to be attained with relatively lighter longitudinals.

RELATION OF USEFUL LIFT AND FUEL LOAD TO GROSS LIFT.

The ratio of useful/gross lift is 35.4 per cent for the *Shenandoah,* and 42.2 per cent for the *Akron,* based on 95 per cent inflation with helium lifting .062 pounds/cubic feet. The relative fuel loads are even more favorable to the *Akron,* amounting to 31.5 per cent of the gross lift in that ship, as compared with only 20.8 per cent in the *Shenandoah.* It is very difficult to determine how much of this improvement should be credited to design, and how much to size. An airship of the volume of the *Shenandoah,* but having the characteristics of the *Akron* with respect to slenderness ratio, type of girders, wiring system, and relative spacing of main frames, but retaining wired main frames and engines in external cars as in the *Shenandoah,* could probably be built with nearly as large a ratio of useful to gross lift as the *Akron,* meeting the same strength requirements as the *Akron,* and with the speed proportionate to $\sqrt{L}$, which would be 64 knots for a ship of the size of the *Shenandoah* stepped down from the *Akron.* If not required to carry airplanes, the ratio of fuel load to gross lift could probably be nearly as high in this hypothetical airship as in the *Akron.*

SPEED AND PROPULSIVE COEFFICIENT.

The designed speed has advanced from 50 knots in the *Shenandoah* to 72 knots in the *Akron.* It is to be noted, however, that the speed of the *Shenandoah* was limited to 50 knots by the strength of the hull. As originally built, she had six engines, with a total horsepower of 1800, although only about 1050 horsepower was required at 50 knots.

The data on power, fuel consumption, endurance, etc., at 50 knots includes the effect of water recovery apparatus, and favors the *Akron,* since a greatly improved type of apparatus is assumed

for that ship. The 50 per cent gain in propulsive coefficient from the *Shenandoah* to the *Akron* is mainly due to reducing the parasite resistance of the cars, water recovery, and tail surfaces. Some gain is attributable to reducing the slenderness ratio from 8.6 in the *Shenandoah* to 7.25 in the *Los Angeles,* and 5.9 in the *Akron;* but probably the resistance is not greatly affected by this change. Only about 6 per cent improvement, or 12 per cent of the actual 50 per cent increase in the propulsive coefficient, is directly due to the greater size of the *Akron.*

The propulsive coefficient is defined by:

$$K = \frac{\rho v^3 V^{2/3}}{550\ hp}$$

Where K is the propulsive coefficient, and is dimensionless for consistent units of the other dimensions. ρ, v and V are as defined under the strength considerations.

When the propulsive coefficient is constant, the total fuel consumption varies as $V^{2/3}$ at a given speed, so that there is a direct gain in endurance proportional to $V^{1/3}$, or 47 per cent advantage for the *Akron* over the *Shenandoah,* directly due to increased size.

SUMMARY.

The fourfold increase in endurance shown by the U. S. S. *Akron* over the *Shenandoah* may be divided between the following principal causes:

(1) Improved propulsive coefficient, 50 per cent gain.
(2) Better specific fuel consumption, 17 per cent gain.
(3) Direct gain from increased size, 47 per cent.
(4) Relatively larger fuel load, 55 per cent.

Total gain equals $1.50 \times 1.47 \times 1.17 \times 1.55 = 4.00$.

Of these four sources of improvement, 12 per cent of the first is due to size, and the remainder to improved design. All of the second is due to better engine design. The third is entirely a matter of size; and the fourth is mainly, but not wholly, due to improved design.

The increased strength is almost entirely attributable to better design.

FUTURE DEVELOPMENT.

The future will probably see some further improvements in design, along the same lines of progress as from the *Shenandoah* to the *Akron,* in respect to stronger girders and reduced secondary stresses; but not very much can be hoped for in these directions. Improving the form of hull and reducing the parasite resistance can hardly proceed much further. In fact, the internal placing of the engines may not be continued; and a reversion to external engine cars, but of much cleaner lines than those of the *Shenandoah* is quite possible, at the cost of a slight increase in parasite resistance. Improving the performance by reducing the number of gas cells has certainly been carried as far as safety permits; and some reversion in this matter is possible. The only direction in which marked structural improvement seems possible is in reducing the weight and complexity of the transverse framing and joints.

Progress in power plant design will undoubtedly continue, including the introduction of engines running on fuel oil with a very low specific consumption.

Increasing the size remains the chief resource of the airship designer seeking for better performance, although the rate of improvement from this cause alone is not so great as is generally supposed.

MODERN MACHINERY FOR THE YANGTZE RIVER GUNBOATS.

By Lieutenant Carl J. Lamb, U. S. N. R., Member.

Many interested marine engineers and naval officers have noted and commented upon Commander Bryson Bruce's complete and excellent description of the U.S.S. *Guam* and U.S.S. *Tutuila* as given in the August, 1928, issue of the Journal of the American Society of Naval Engineers. Without doubt, all members of the Society are indebted to Commander Bruce for his faithful and accurate description and for the data which he has given us, particularly that which applies to the trials of the two ships. As a matter of fact, this article would not have been possible, as a constructive engineering presentation, without reference to the very accurate data given by Commander Bruce.

At the time the river gunboats were contracted for, there was not available any other type of propulsion machinery which would have been more adapatable than the reciprocating engine, which was the type selected. Quite recently, however, great strides have been made in a form of steam propelling machinery which is simpler, more reliable, weighs less, takes up less space and uses much less steam and fuel than the reciprocating engine and as a result of its remarkable performance it is being widely acclaimed by marine engineers.

The type referred to is the overhung wheel, combined re-entry single row marine geared turbine, invented and developed by Mr. Henry F. Schmidt, twice prize essay winner of this Society.

At the present time ten such units are in service or are being installed, as follows:

2—650-S.H.P. units, 60 per cent astern power, in a submarine chaser, 300 pounds steam pressure, 50 degrees F. superheat and 27 inches vacuum.

1—1250-B.H.P. unit, no astern power, driving a 30-inch dredge pump, 150 pounds steam pressure, 0 degree F. superheat and 26 inches vacuum.

1—1000-S.H.P. unit, 100 per cent astern power, tug *Harry B. Williams,* 300 pounds steam pressure, 200 degrees F. superheat and 28 inches vacuum.

4—800-S.H.P. units, 60 per cent astern power in two canal barges, 400 pounds steam pressure, 200 degrees F. and 27 inches vacuum.

2—550-B.H.P. units, no astern power, driving two 20-inch dredge pumps, 200 pounds steam pressure, 0 degree F. superheat and 28 inches vacuum.

Since this type of geared turbine was not fully developed in 1925 when the River Gunboats were authorized and was not being offered for such service, it is not in any way the intent of this article to imply that proper machinery was not selected, or that any other form should have been chosen at that time.

Because of the vast and broad experience available today, as to design, construction and operation of practically all sizes, types and powers of land, marine, dredging and naval turbines covering the entire field of power generation, it is doubtful if Commander Bruce still feels that "geared drive would probably give serious operating difficulties, due to the vibrating pulsations in power being transmitted by propellers on account of their location near the surface and the turbulent water of the gorges." Certain it is that other experienced turbine engineers could not agree with such an opinion today. Just consider, for example, a few of the varied and difficult requirements served by geared turbines at the present time:

A—3300-B.H.P. geared turbine driving a two stage, two crank, 550 pounds pressure, gas compressor. The cranks are located on each end of the low speed gear shaft, set 90 degrees apart. This unit was installed in 1914 and has been running ever since.

B—A triple reduction geared turbine side paddle wheel river steamer that has been running since 1925 on the River Rhine in Germany.

C—The 1200-S.H.P. double reduction geared low pressure turbine of constant torque running on the same shaft with a

2800-H.P., triple expansion reciprocating engine of variable torque, aboard the S.S. *Susan V. Luckenbach* and over 100 such installations built by three builders abroad.

D—The British Navy has had at least one geared turbine River Gunboat operating on the Yangtze for some time.

E—The 1400-B.H.P. geared turbine driving a 20-inch dredge pump aboard the suction dredge *New Jersey*. This unit replaced a 750-B.H.P. reciprocating engine, going into the same space, weighing less and delivering twice the power on less fuel. In twenty months, although the pump has been completely wrecked by stones, and the drive shaft sheared, there has been no money spent for repairs, overhaul or maintenance, on the geared turbine. Only one man is on watch in the engine room at a time.

F—The ten units cited earlier.

The geared turbine is inherently a simple piece of apparatus, with few moving parts and lends itself more easily to a design that has its first critical speed away above its running speed, than other prime movers. In addition the geared turbine can be properly balanced far easier than any form of reciprocating drive, be it steam or Diesel.

In examining the trial trips data given for the *Guam* and *Tutuila* two things strike one immediately. The fuel consumption rates of 2.16 and 2.04 pounds of fuel per I.H.P. hour respectively for the *Guam* and *Tutuila* at 1395 I.H.P. and 1360 I.H.P. are amazingly high and are equivalent to 2.4 and 2.2 pounds of fuel per hour on a S.H.P. basis. These high fuel rates may be partly explained by the exceedingly low feed temperatures of 156 degrees F. and 150 degrees F. as given, although it is hard to understand why the feed temperatures are so low with such a considerable amount of auxiliary exhaust steam being available from so many reciprocating auxiliaries.

In view of the fact that a more efficient, more simple and lighter weight type of main engine drive with improved auxiliaries is now available for such service and since it would result in a marked decrease in fuel consumption, a considerable saving in space and a reduced displacement, the writer has taken the liberty of assuming that the *Guam* and *Tutuila* were both modernized with this new type of engine and with improved auxiliaries and offers the

following comparative data of the boats as they are now equipped and as they would be after re-engining:

GENERAL DATA.

	As Described by Commander Bruce	As Proposed
Length between perpendiculars, feet and inches	150—00	150—00
Length overall, feet and inches	159—03¼	159—03¼
Length on load water line, feet and inches	149—09	149—09
Beam moulded, feet and inches	27—00	27—00
Beam extreme, feet and inches	27—11	27—11
Draft, displacement fresh water, feet and inches	5—3	4—8½
Displacement on displacement draft tons	387	340
*Fuel oil carried on displacement draft tons	26.	15.
*Fuel oil bunker capacity, tons	77.48	44.
*Fuel oil bunker capacity, tons	6 (for new ships)	4

While no great objection can be raised as to the steam anchor gear and steering gear, it is believed that weights could be considerably reduced, efficiency improved and equal or better reliability obtained by the use of an electric anchor gear and a turbine driven hydraulic steering gear (similar to the one installed on the 1000–S.H.P. Great Lakes Tugboat *Harry B. Williams*).

Under the contract requirements, which were easily met as shown by the result of the trials, the *Tutuila* was able to cruise 1000 miles at 10 knots on a total consumption of 34.75 tons. This total consumption would become 20 tons if she were fitted with the overhung wheel type of geared turbine and certain modern auxiliaries and were to use an easily obtained feed temperature of 230 degrees F.

Figure 1 shows the modernized engine room arrangement proposed with modern auxiliaries when justified.

Figure 2 shows what could be done with all new and modern main and auxiliary machinery in designing an engine room for new River Gunboats.

On the basis of modernizing the existing main machinery plant, new boilers would hardly be justified, but for a new ship they undoubtedly would be, since their weight, cost and size would be considerably less.

* Based on same cruising radius for either form of drive.

Figure 3 shows suggested boiler room arrangement for a new geared turbine vessel.

An examination of Figures 2 and 3 shows that the overall length of engine room and fire room is 22 frame spaces, or 8 frame spaces (16 feet) shorter than the present overall length of the engine room and fire room, which is 60 feet (30 frame spaces).

Since the existing engine room and fire room extend through the main and upper decks, where they take up a total of 62 feet or 26 frame spaces, it is evident that with the same overall length and beam, machinery arrangements as shown in sketches 2 and 3 not only give 16 feet additional space in the hull, but also give 16 more feet additional space on the main and upper decks.

With steam conditions of 250 pounds pressure at the boiler, dry and saturated, the total steam consumption for all purposes of approximately 35,000 pounds per hour, at 14.5 knots, and 1260 shaft horsepower can be reduced by modernization to 20,000 pounds per hour for all purposes, which would reduce the fuel rate from 2770.5 pounds of fuel oil per hour to 1590 pounds of fuel oil per hour, or from 2.2 pounds of oil per S.H.P. hour, for all purposes, to 1.26 pounds of oil per S.H.P. hour for all purposes.

For a new vessel, with steam conditions of 300 pounds boiler pressure, 200 degrees F. superheat and 28 inches vacuum, the overall fuel consumption could be brought to one pound or under of fuel oil per S.H.P. per hour for all purposes and the displacement tonnage would be further reduced.

ADVANTAGES OF MODERNIZATION.

There would be no good reason for proposing or considering such modernization unless definite advantages could be obtained. While the writer may be wrong, he believes that the following advantages would justify re-engining the existing River Gunboats with modern main and auxiliary machinery or the use of such machining for any similar future vessels.

1. Reduced displacement and draft.
2. Greater space available for living and military purposes.
3. Greatly reduced fuel costs.
4. Reduced lubricating oil costs.

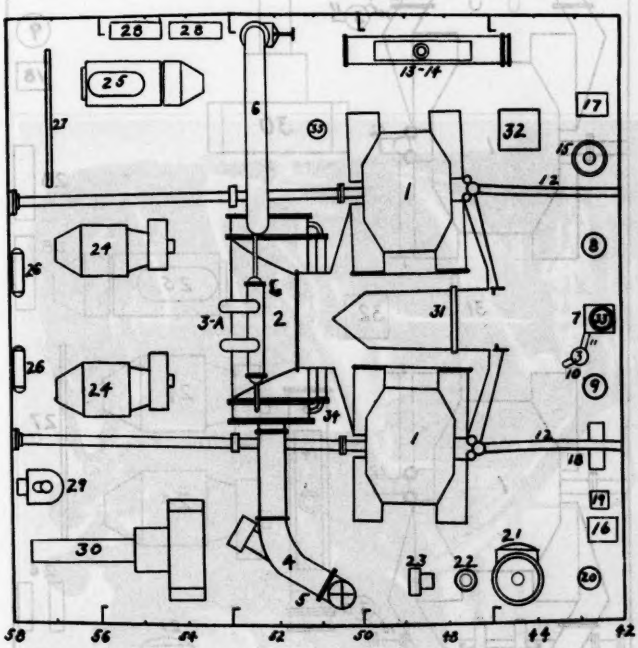


Fig. 1

* Indicates existing equipment to remain.

1—Main turbines, gears and thrusts.
2—Main condenser, single pass, welded steel.
3—Main condensate pump, vertical motor driven.
3—A—Main air ejectors.
4—Main propeller type circulating pump, turbine driven.
5—Main injection.*
6—Overboard discharge.
7—Feed and filter tank.*
8—Vertical turbine driven, new main feed pump.
9—Auxiliary feed pump.*
10—Condensate pump suction.
11—Condensate pump discharge.
12—Main steam line.
13—Auxiliary condenser.*
14—Auxiliary air and circulating pump.*
15—Grease extractor.*
16—General service pump, starboard.*
17—General service pump, port.*
18—Motor driven, general service pump.*
19—Fresh water pump.*
20—Sanitary pump.*
21—Evaporator.*
22—Distiller.*
23—Evaporator feed and distiller circulating pump.*
24—25-K.W. Westinghouse Turbo-Generator.*
25—10-K.W. Oil Engine Generator.*
26—I.C. Motor Generator.*
27—Main switchboard.*
28—Searchlight rheostat.*
29—Sensitive drill.*
30—Combination lathe.*
31—Gage board.
32—Log desk.*
33—Feed heater.*
34—Oil cooler.
35—Oil purifier.

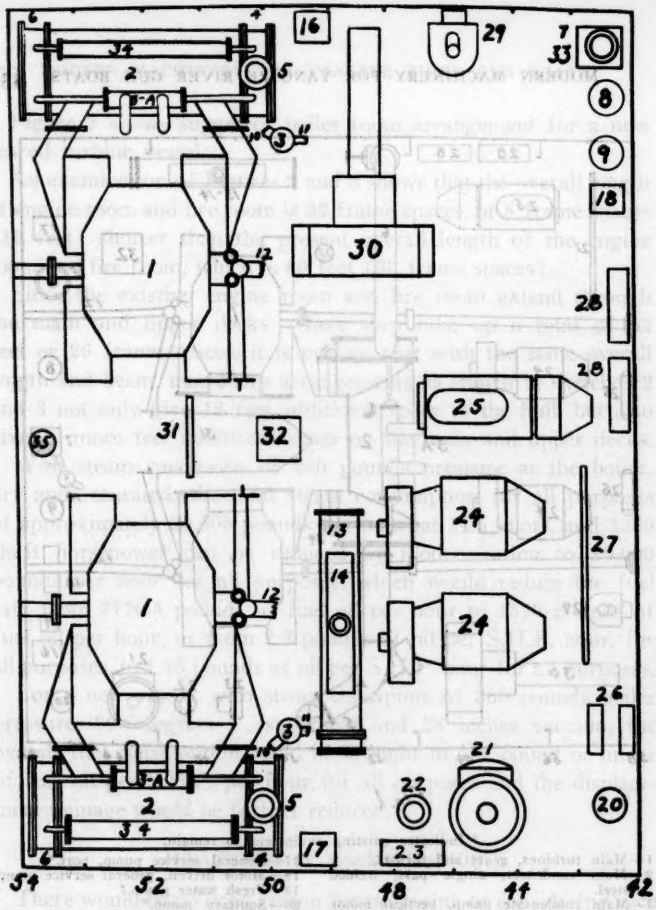
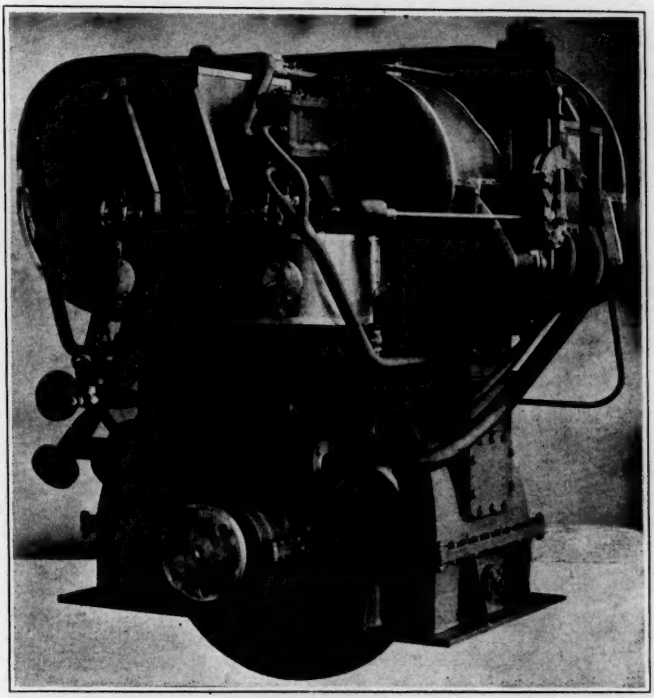
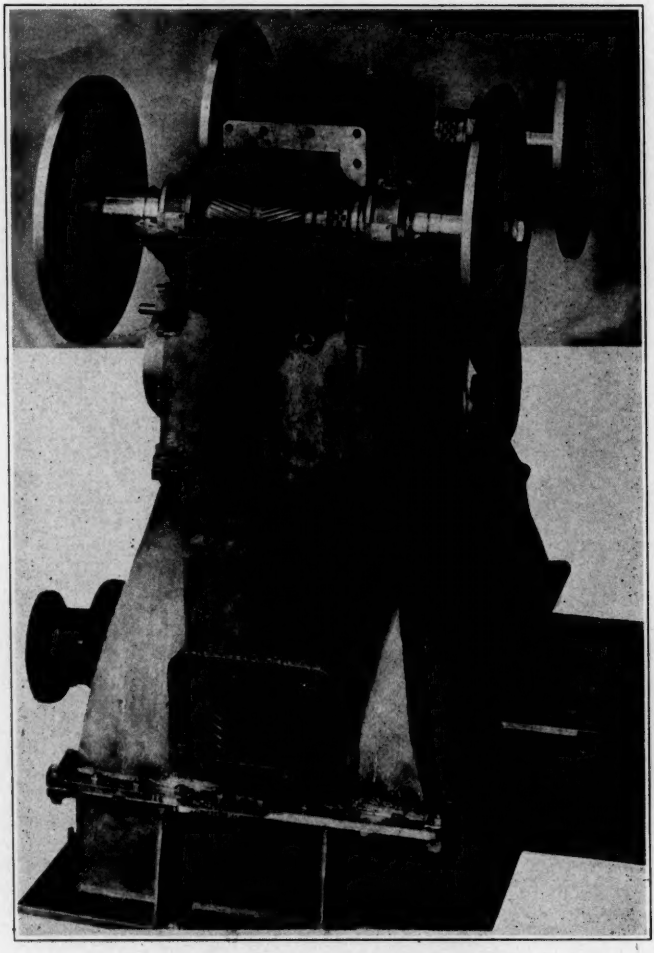


Fig. 2

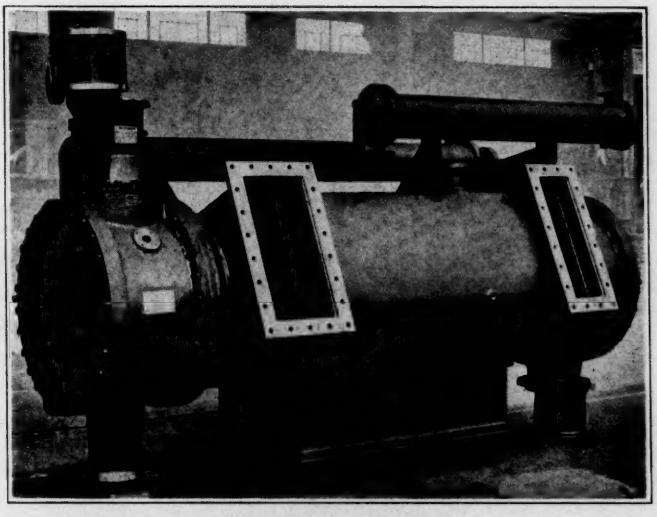
1—Main turbines, gears and thrust, also steam strainers and throttles.
2—Main condensers, single pass with circulating pumps in water boxes.
3—Main motor driven condensate pumps.
4—Main circulating pumps, turbine driven, mounted in condenser water boxes.
5—Main injections.
6—Main overboard discharges.
7—Feed and filter tank.
8—A main and auxiliary vertical turbine driven, 3 stage boiler feed.
9—Pumps.
10—Condensate pump suctions.
11—Condensate pump discharges.
12—Main steam lines.
13—Auxiliary condenser.
14—Auxiliary motor driven circulating and condensate pump, with air ejector.
15—Grease extractor over No. 18.
16—17—Starboard and port turbine driven general service pumps.
18—Motor driven, general service pump.
19—Motor driven fresh water pump.
20—Motor driven sanitary pump.
21—Evaporator.
22—Distiller.
23—Motor driven evaporator feed and distiller circulating pump.
24—2—25-KW turbine generators.
25—1—10-KW gasoline engine generators.
26—1—C motor generators.
27—Main switchboard.
28—Searchlight rheostat.
29—Sensitive drill.
30—Combination lathe.
31—Overhead gage board.
32—Log desk.
33—Feed heater.
34—Oil coolers, mounted on condensers.
35—Oil purifier.



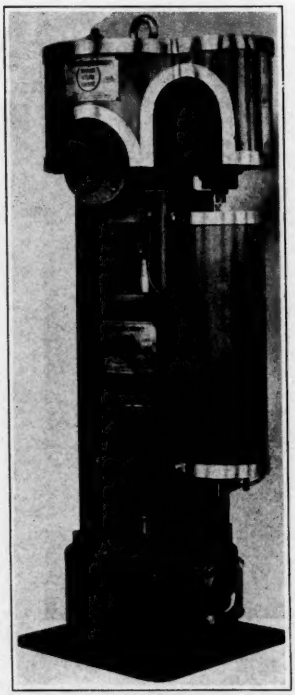
A.—The New Type of Geared Compound Impulse Turbine. 800 S.H.P. at 250 R.P.M. with Steam at 385 Pounds Gage Pressure, 200 Degrees F. Superheat and 27-inch Vacuum. This Unit Has 65 Per Cent Astern Power.



B.—A View of the Same Geared Turbine Unit with the Cylinders Removed.

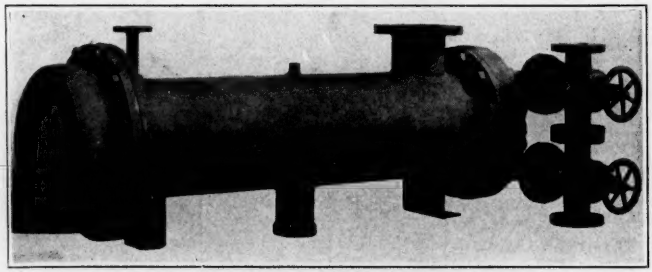


C.—A 520-Square-Foot Condenser for the Above Unit. The Turbine Driven Propeller Type Circulating Pump Is Located in the Water Box. The Air Ejector Condenser and Oil Cooler Are Mounted on the Main Condenser and Results in a Saving of Foundation Floor Space, and Piping. The Condenser Itself Is on Flexible Steel Supports Which Eliminate the Necessity of Expansion Joints in the Exhaust Trunk.

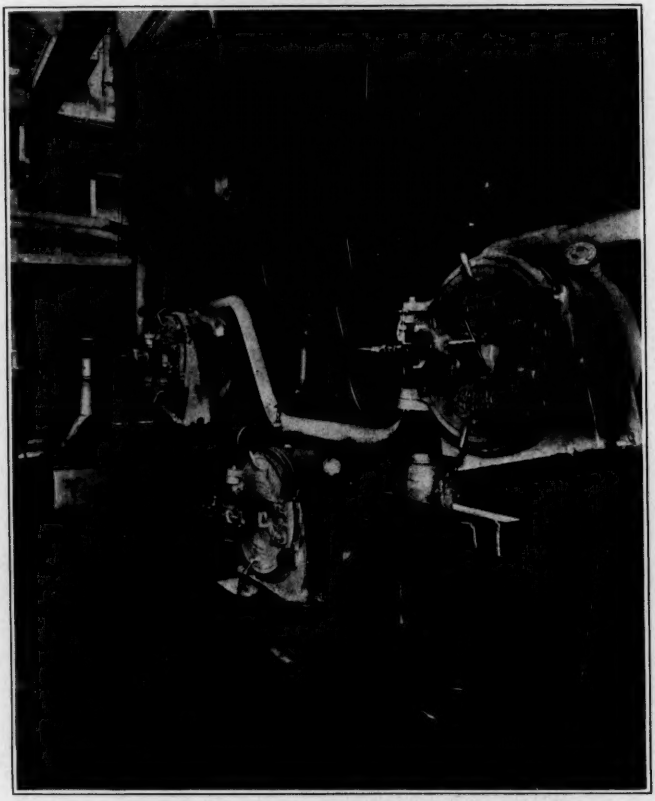


D.—A Modern Vertical Turbine Driven Boiler Feed Pump that Weighs Only 750 Pounds and Delivers 90 G.P.M. at 550 Pounds Pressure.

E.—A Modern Vertical Turbine Driven Condensate Pump Which Weighs Only 550 Pounds and Delivers 60 G.P.M. Against a Total Head of 60 Feet.



F.—A Modern Feed Water Heater.



G.—Installation View of a Modern Vertical Turbine Driven Propeller Type Forced Draft Blower in a Tugboat Fire Room.

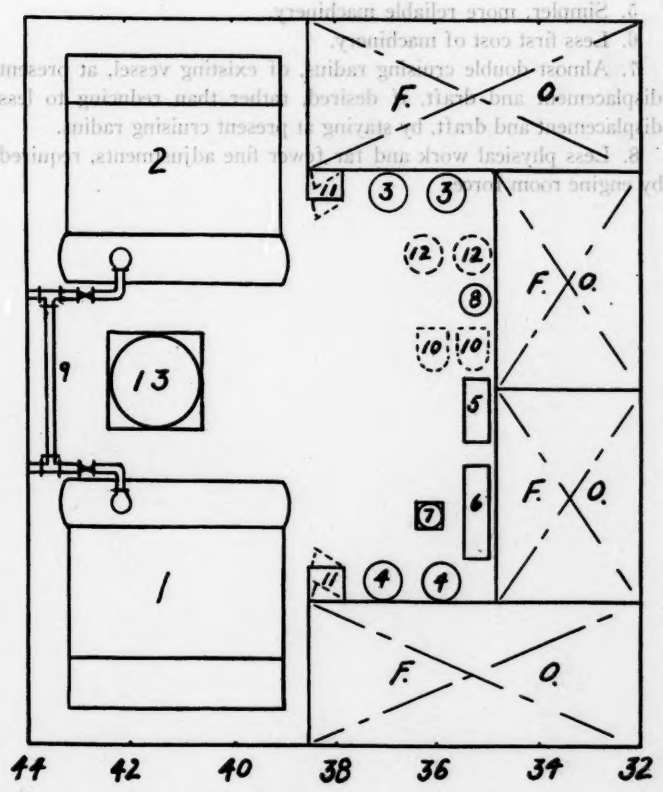


Fig. 3

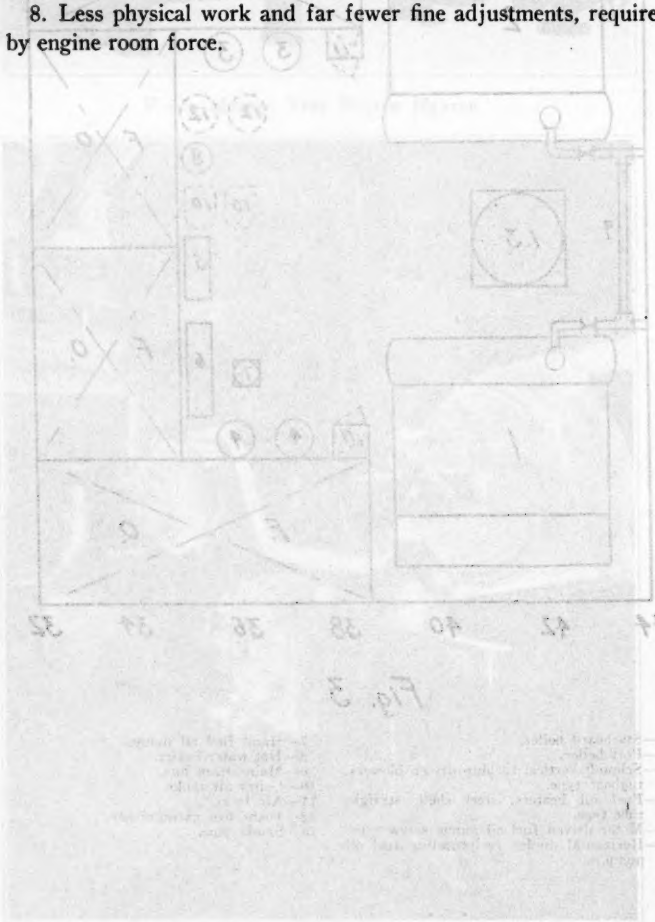
1—Starboard boiler.
2—Port boiler.
3—Schmidt vertical turbine driven blowers, tugboat type.
4—Fuel oil heaters, steel shell, straight tube type.
5—Motor driven fuel oil pump, screw type.
6—Horizontal duplex reciprocating fuel oil pump.
7—Hand fuel oil pump.
8—Hot water heater.
9—Main steam line.
10—Galley air tanks.
11—Air locks.
12—Foam fire extinguisher.
13—Smoke pipe.

5. Simpler, more reliable machinery.

6. Less first cost of machinery.

7. Almost double cruising radius, of existing vessel, at present displacement and draft, if desired, rather than reducing to less displacement and draft, by staying at present cruising radius.

8. Less physical work and far fewer fine adjustments, required by engine room force.



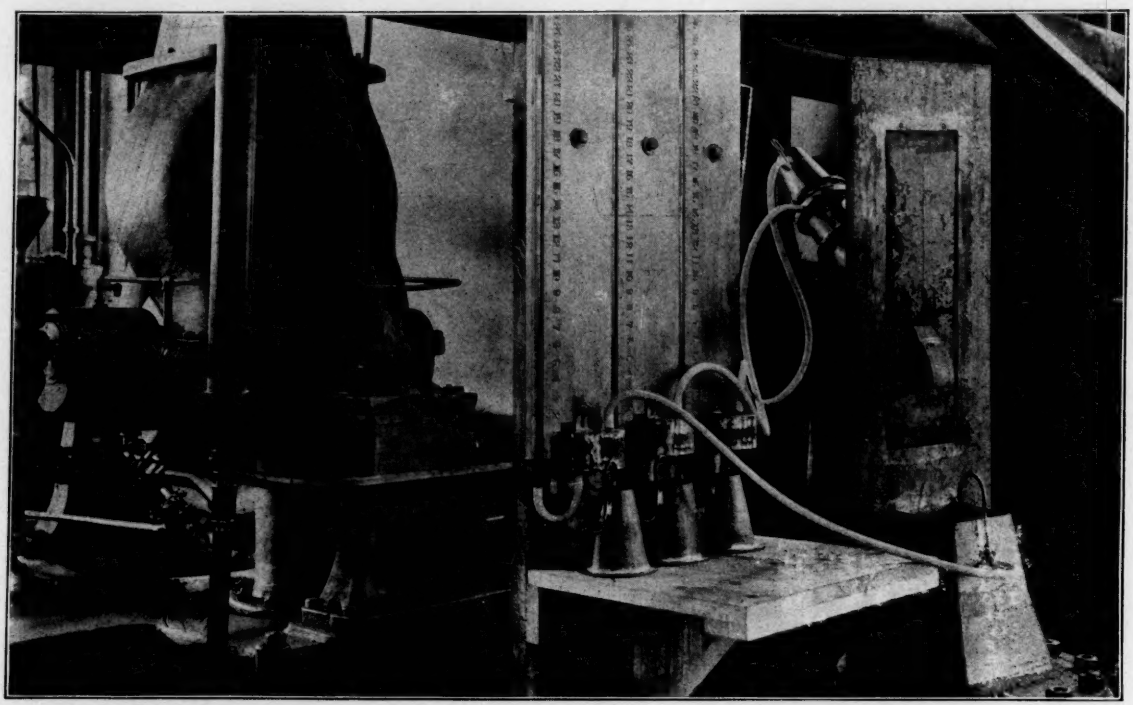


FIG. 1.

TEST ON A ONE-QUARTER SCALE MODEL SCOOP ON THE U.S.S. *WELBORN C. WOOD* AND PREPARATORY LABORATORY EXPERIMENTS.

By Henry F. Schmidt, Member, and Captain O. L. Cox, U. S. N., Member.

Since the publication of the paper "Theoretical and Experimental Study of Condenser Scoops," published in the February, 1930, issue of the Journal of the American Society of Naval Engineers, considerable further work has been done in the laboratory.

A consideration of the results of the tests previously reported and particularly a study of the characteristics of flow into the standard form of scoop, led to the belief that the lip, commonly employed, might well be omitted. To recall the test arrangement employed, Figure 15 from the article referred to is herewith reproduced as Figure 1.

The stream corresponding to the flow of water past the ship is produced by the square nozzle in the lower portion of the photograph and the scoops to be tested are inserted in a removable plate, as shown. The velocity head of the approaching stream, the static pressure, and the total head produced by the scoop are registered on the three gages shown. From the total head and the size of the nozzle, the volume is determined.

Referring to Figure 2, showing the observed stream line flow in a standard type of scoop, it is observed that the fluid enters the forward portion of the scoop and that some fluid actually comes out of the lip. This indicates the possibility that the lip may be dispensed with entirely. Accordingly, tests were undertaken to determine the effect of cutting back the lip, as indicated by the sketch in Figure 3. First, the scoop was tested with the entrance edge of the lip making an angle of 90 degrees with the hull, that is, the complete elbow was external to the skin of the ship. Then, the lip was cut off, as indicated, at an angle of 75 degrees, 60

degrees, 45 degrees and 22.5 degrees and finally the lip was removed entirely. The results of these tests show that the characteristic is entirely changed, but as is seen from the intersection of the curves, at the larger volumes, the performance at approximately normal capacity or above, is substantially the same with or without the lip. The normal capacity of the original scoop was 335 cubic feet per minute but this increased when the lip was entirely removed, to about 360 cubic feet per minute because of the increased area, consequently for comparison, the curves without the lip should be shifted towards zero 30 cubic feet. The first tests run were with a scoop having a 45-degree elbow, and further tests were made with a scoop having a 60-degree elbow. Of course the general design of the scoop, originally laid out with a lip, was not favorable when the lip was removed, so new scoops were made, designed without the lip. Two curves of performance of these newer scoops are shown on the lower part of Figure 3.

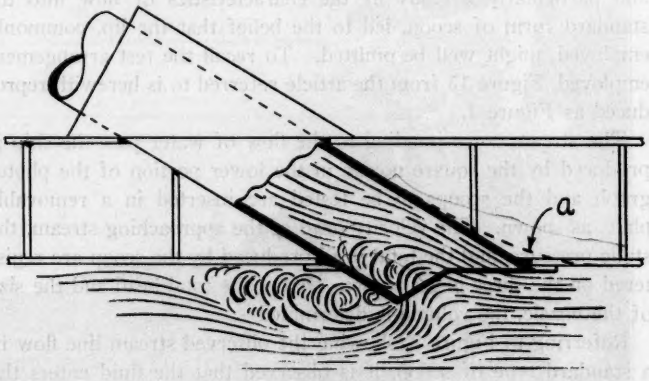


FIG. 2.

The design of the scoop was then modified and splitters were introduced to determine their effect.

In order to determine the stream line form of the flow entering the lipless scoop, a 1/16-inch thick polished copper sheet was soldered onto the face of the plate carrying the scoop model as shown in Figures 4 and 5, Figure 4 showing a side elevation and

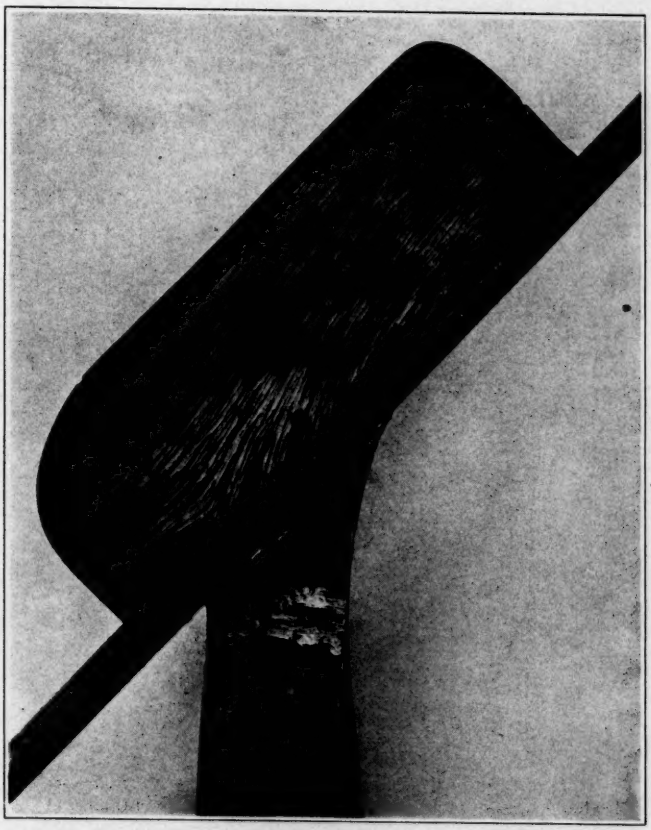


FIG. 4.

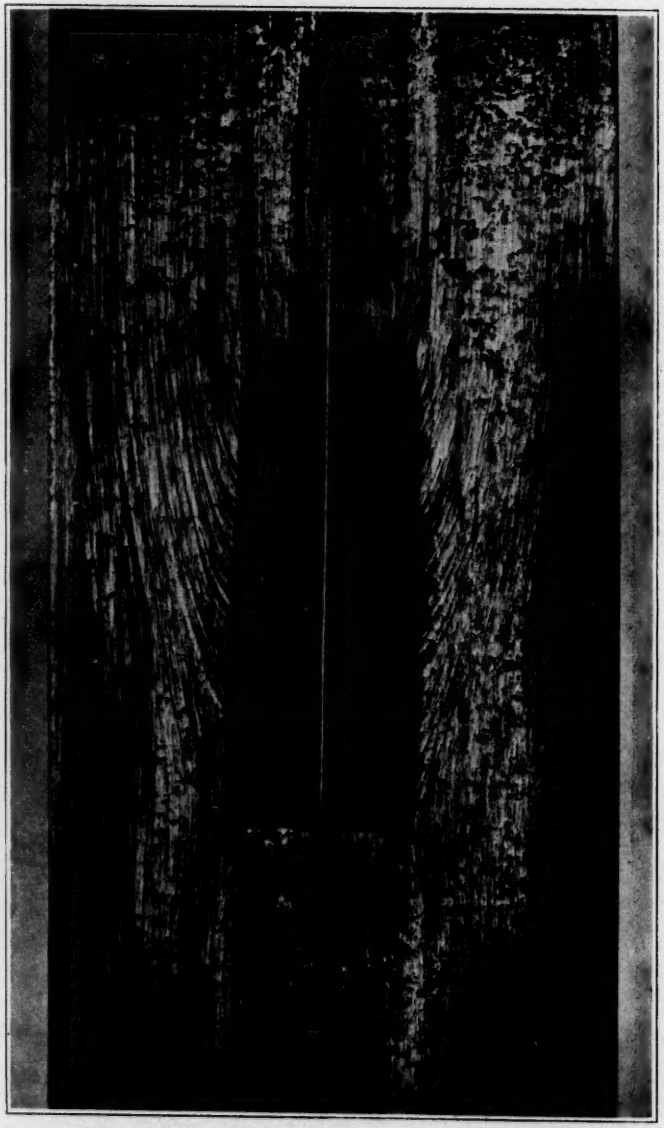


FIG. 5.

Figure 5 a plan view. The copper plate was painted with lamp black and cylinder oil.

These stream lines were obtained at a flow of approximately 120 per cent of normal capacity, that is, the velocity of the air entering the scoop was 20 per cent higher than the velocity of the approaching stream. It is observed in both the side and plan views, that the stream lines converge toward the opening, which must necessarily be the case, since the pressure at the entrance of the scoop is very materially lower than that of the surrounding atmosphere, consequently, the flow into the inlet is similar to that in a nozzle. Particular attention is called to the fact that in spite

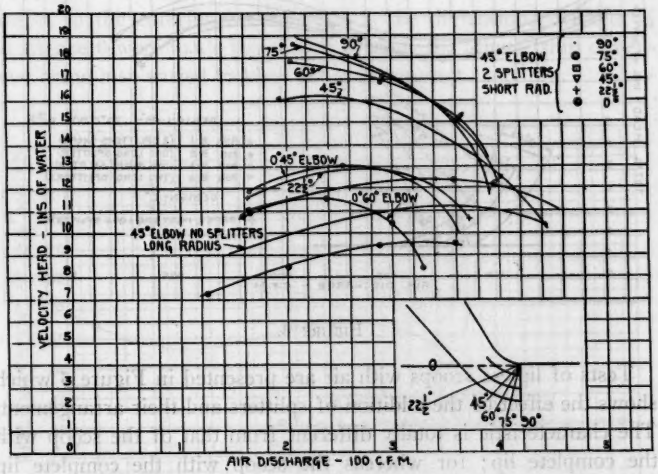


FIG. 3.

of the absence of a lip, the air enters in exactly the same stream line form which it would have with the 90-degree or complete elbow. In this particular case, the entrance of the scoop is 1⅞ inches high and an examination shows the stream lines within 1⅞ inches out from the plate just enter the inlet. The flow into the inlet causes a change in the direction of the stream lines and the photographs show that the air beyond 1⅞ inches from the plate which does not enter the scoop is deflected away from the

supporting plate. Also, in the plan view, it will be observed that the stream lines converge toward the scoop from the side and just beyond the scoop inlet they diverge owing to the impact pressure of the stream lines impinging on the plate from above. It is thus seen that the lipless scoop operates in the same manner as though the lip were present.

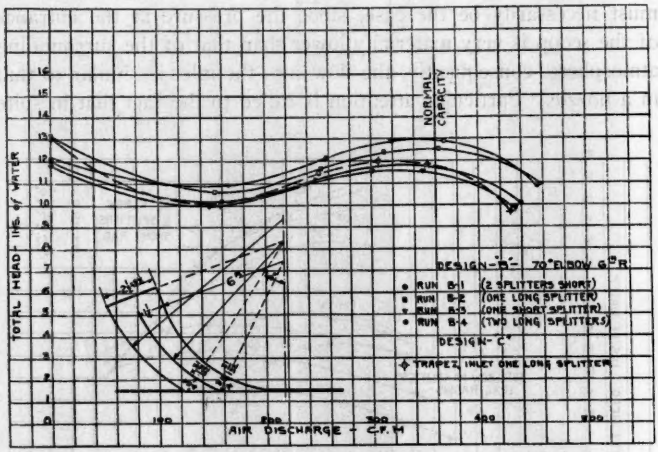

TOTAL HEAD - INS. of WATER
NORMAL CAPACITY
DESIGN-"B"- 70°ELBOW 6"R
RUN B-1 (2 SPLITTERS SHORT)
RUN B-2 (ONE LONG SPLITTER)
RUN B-3 (ONE SHORT SPLITTER)
RUN B-4 (TWO LONG SPLITTERS)
DESIGN- C"
TRAPEZ INLET ONE LONG SPLITTER
AIR DISCHARGE - C.F.M.

FIGURE 6.

Tests of lipless scoops with air are presented in Figure 6 which shows the effect of the addition of splitters and their arrangement. The characteristic is totally different from that of the scoop with the complete lip; for whereas the scoop with the complete lip shows a continuously rising characteristic with decrease of volume when not fitted with strainer plates the characteristic with strainer plates is an increase in pressure with decrease of volume to a point slightly below normal capacity and then a decrease in pressure with decrease in volume; whereas the lipless scoop shows an increase in pressure up to normal capacity, then a decrease in pressure to approximately ¼ normal capacity, and then an increase in pressure up to shut off. In Figure 6, the curve for design "C" with trapezium inlet refers to the shape of the inlet when viewed inboard or in plan. The inlet was made wider athwartship at the entering

edge and narrower at the after edge, with the thought that this would improve the performance above normal capacity as it would avoid the entering of so much fluid from the sides as indicated in Figure 4. This fluid entering from the sides in Figure 4 does not have the same velocity along the axis of the scoop as the fluid entering as in Figure 3 along the more central portion of the inlet and consequently if this side flow could be eliminated, the performance of the scoop would be improved. As indicated by the test, however, for some reason difficult to discover, this did not prove to be the case.

A number of designs, a, b, c, d, etc., were tested, all of the same radius of curvature and the same inlet area taken at right angles to the entering stream, that is the dimensions marked ¾ in Figure 6 were made larger and smaller, the athwartship dimension being also varied to keep the area constant.

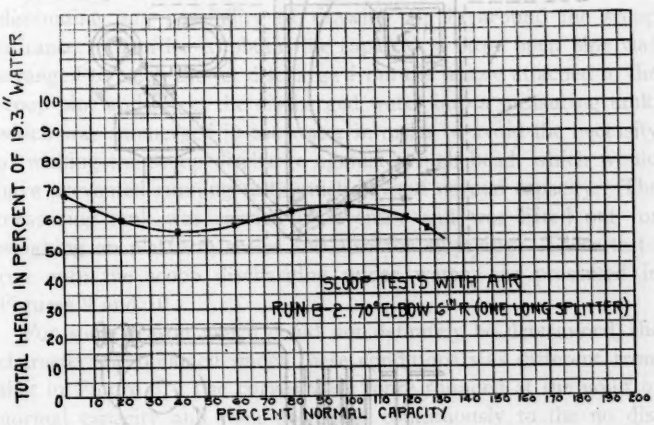


FIG. 7.

Run B 2. with one long splitter most nearly corresponds to the ¼ scale model which was tested on the *Wood* and for ease of comparison, this curve has been reduced to a percentage basis in Figure 7.

The total head and volume of discharge are both expressed on a percentage basis of pressure and capacity at normal flow and

that makes this curve independent of the medium and the size of the scoop.

As reported in the previously published paper on scoops, the scoops with a complete lip were tested with water as well as air. In the case of the scoops with a complete lip the tests with water checked exactly with the test with air when reduced to a percentage basis. A similar set of tests with the arrangement as in Figure 1, except that the nozzle and scoop were inverted, were employed for conducting tests of the lipless scoop with water. These tests check the tests with air up to substantially the point of normal capacity. Above this point, however, the scoop broke down completely and it was impossible to obtain any volume measurements exceeding normal capacity, which was attributed to the

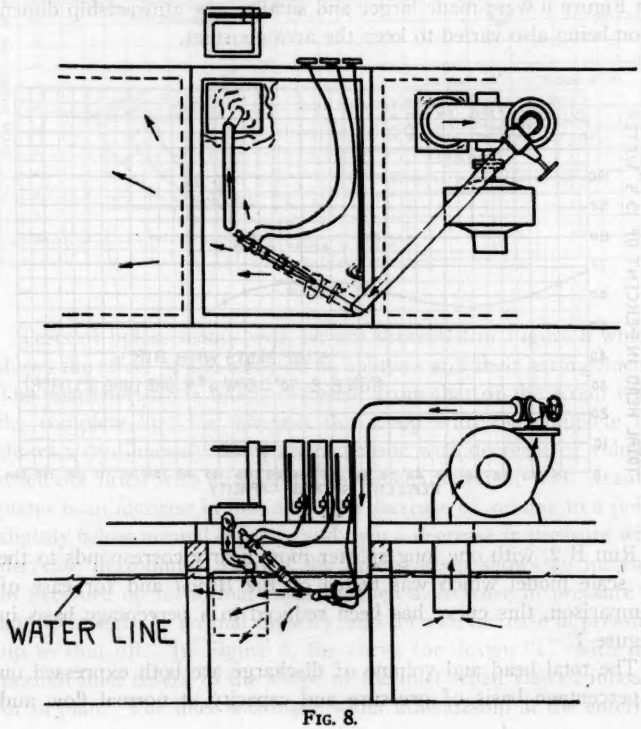


FIG. 8.

fact that, as the scoop entrance was approximately 2 inches wide and the jet of water only 5 inches wide, when normal capacity was exceeded and the pressure at the entrance to the scoop fell considerably below atmospheric, air was drawn into the scoop inlet which caused a complete breakdown of the divergent tube, which was indicated by the fact that, under these conditions, a high velocity jet came out of the nozzle but did not completely fill it, showing that the stream had broken away from the walls of the divergent tube and consequently, as there was no conversion of velocity into pressure, it was, of course, impossible to exceed normal capacity.

To avoid this condition, a set up was made as shown in Figure 8, which gives a plan and elevation of the arrangement adopted. In this case, the discharge nozzle and plate for holding the scoop illustrated in Figure 1, was submerged under water, with the scoop projecting up through the water level and the plate in a horizontal position with the scoop entrance under the plate, thus eliminating any possibility of drawing air in around the scoop entrance. In order to obtain the capacity, a large open pipe was arranged to catch the jet discharge from the nozzle attached to the scoop pipe and direct the discharged water into a measuring tank, which was submerged in the water in order to avoid the necessity of making the scoop discharge against a high head, which would have prevented running tests much beyond normal capacity. The measuring tank was carried by a crane and was lifted out for weighing on platform scales and also for emptying. These tests, run with the scoop discharging under water, are presented in Figures 9 and 10.

For some reason which could not definitely be determined, the characteristic obtained under these conditions was different from that in Figure 6. The curves were more rounded at the point of normal capacity and then decreased continuously to the no discharge condition instead of rising, as in Figure 6, also, reduced to a percentage basis, the maximum total head is about 10 per cent lower than the test run either with air,—or with water with the water jet discharging in air instead of under water. Whether this was caused by entrainment of the circulating water lowering the jet velocity, or some other hydro-dynamical reason, is not known. Theoretically, as the air jet is surrounded by air, which is a

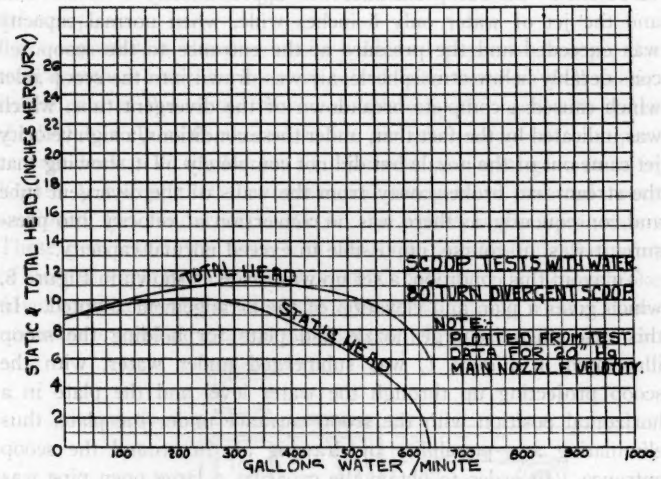

STATIC & TOTAL HEAD (INCHES MERCURY)
0
2
4
6
8
10
12
14
16
18
20
22
24
26
TOTAL HEAD
STATIC HEAD
SCOOP TESTS WITH WATER
80° TURN DIVERGENT SCOOP
NOTE:-
PLOTTED FROM TEST
DATA FOR 20" Hg.
MAIN NOZZLE VELOCITY.
100
200
300
400
500
600
700
800
900
1000
GALLONS WATER / MINUTE

FIG. 9.

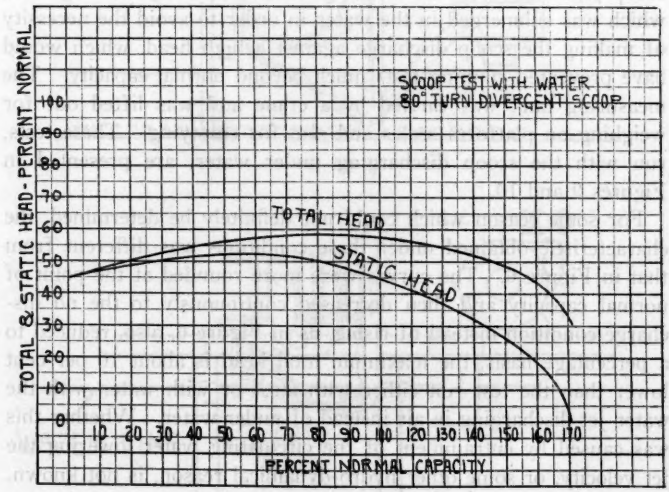

TOTAL & STATIC HEAD - PERCENT NORMAL
0
10
20
30
40
50
60
70
80
90
100
SCOOP TEST WITH WATER
80° TURN DIVERGENT SCOOP
TOTAL HEAD
STATIC HEAD
10
20
30
40
50
60
70
80
90
100
120
130
140
150
160
170
PERCENT NORMAL CAPACITY

FIG. 10.

medium of the same density as the jet, and as this also applies to the jet of water discharging under water, it would seem reasonable to assume that the entrainment effect, expressed as a percentage, should be the same in both cases, and would therefore not account for the difference as observed. As will be seen later, the model scoop, as tested on an actual ship, shows the same general characteristic as the tests with air and the values for total head also substantially check the tests with air.

Tests were also conducted with water to find the pressure drop across the elbow and propeller pump, as was done in the original experiments with air. These tests are not directly comparable, however, since the original elbow was not fitted with a splitter or vertical division plate, as shown in Figure 22. The test arrangement for obtaining the pressure drop is shown in Figure 11. The

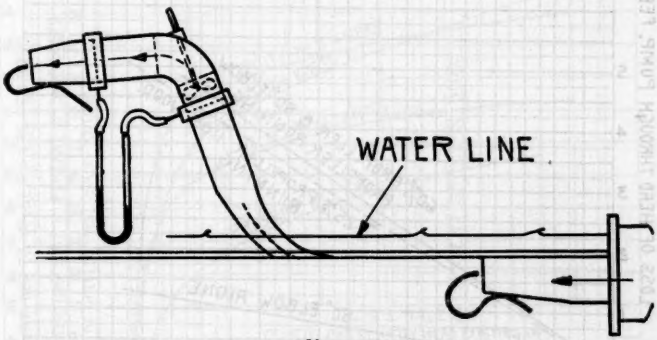


FIG. 11.

flow through the pump was obtained by measuring the discharge through the nozzle, the flow being increased by increasing the velocity of the jet passing the scoop; the pressure drop was measured directly by means of a differential gage, as shown. The propeller employed for these tests had a projected area ratio of 60 per cent, a pitch ratio at the leading edge, of .75, and a pitch ratio at the trailing edge of 1.25. The results of these tests are shown in Figure 12, which also shows the observed pressure drop across the pump on the U.S.S. *Wood*. In the case of the model test, the propeller ran free without having to drive a motor, whereas on

the *Wood,* the 25 per cent area ratio propeller had to drive the motor and consequently ran at a relatively lower speed and introduced a relatively high loss. As the resistance offered by the propeller decreases with increase of speed, it is obvious that a full size pump and elbow would have considerable less loss than a model, since the friction and rotation losses of the motor and bearings will be more or less in proportion to the size of the model, whereas the available energy to produce rotation, at a given velocity, is proportional to the square of the dimensions, and consequently the full size model would run proportionally faster and offer less

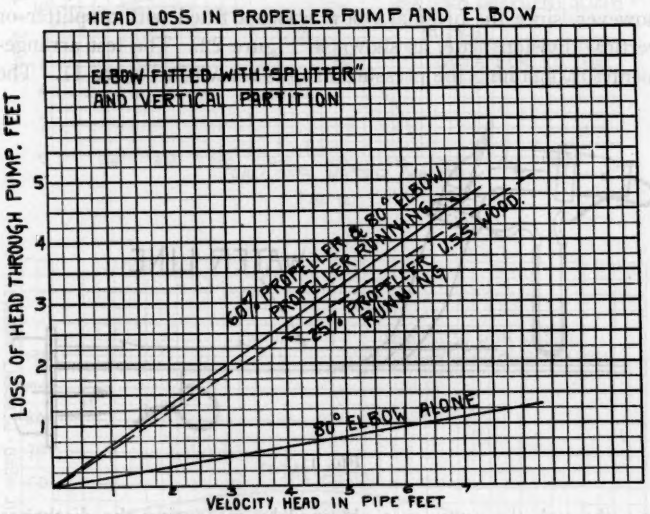


FIG. 12.

resistance. This accounts for the difference in resistance of the model scoop running free with a 60 per cent propeller, and a larger pump, having to drive the motor with a 25 per cent propeller. This indicates the desirability of still further tests, as it may be found that a propeller more efficient as a pump, having say a projected area ratio of 35 or 40 per cent, would also offer less pressure drop than a 25 per cent propeller because being a more efficient turbine it would rotate at a higher velocity when idling.

In Figure 13 the relative velocity of the water, with respect to the ship and the observed corresponding velocity head at different distances from the skin of the ship, are shown as observed by readings taken by means of a search tube on the U.S.S. *Raleigh.* These velocities and pressures represent those existing at the point of measurement. In order to make this data available for scoop calculations, it is necessary to know the mean velocity and the mean velocity head of the water in the layer which effects the scoop. In order to obtain this average velocity and velocity head, the curves of Figure 13 were integrated, obtaining the area under the curves for different distances from the skin of the ship and from those areas under the curves, the mean velocity and velocity head were calculated and plotted in Figure 14.

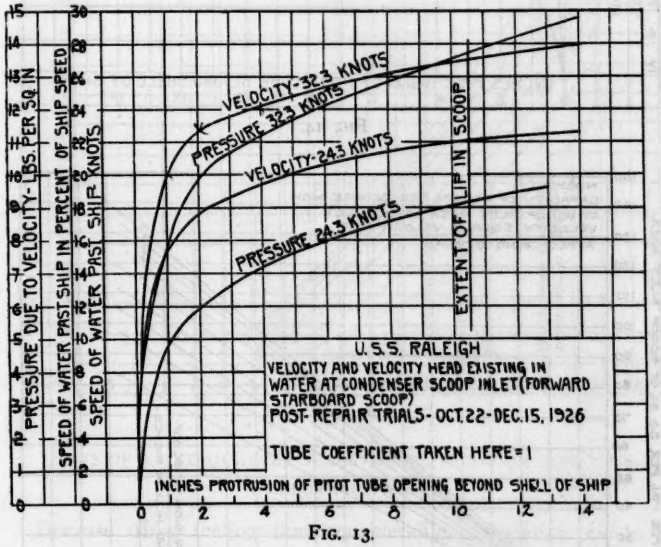

PRESSURE DUE TO VELOCITY-LBS. PER SQ IN
SPEED OF WATER PAST SHIP IN PERCENT OF SHIP SPEED
SPEED OF WATER PAST SHIP KNOTS
VELOCITY-32.3 KNOTS
PRESSURE 32.3 KNOTS
VELOCITY-24.3 KNOTS
PRESSURE 24.3 KNOTS
EXTENT OF LIP IN SCOOP
U.S.S. RALEIGH
VELOCITY AND VELOCITY HEAD EXISTING IN WATER AT CONDENSER SCOOP INLET (FORWARD STARBOARD SCOOP)
POST-REPAIR TRIALS-OCT. 22-DEC. 15, 1926
TUBE COEFFICIENT TAKEN HERE=1
INCHES PROTRUSION OF PITOT TUBE OPENING BEYOND SHELL OF SHIP

FIG. 13.

As the data is only given for two speeds, the curves in Figures 15 and 16 were prepared. The curves for Figure 15 are based on the mean velocity for various protrusions of the scoop and show the capacity in gallons per minute per square inch of scoop inlet area for various protrusions of the lip, or distance to which the

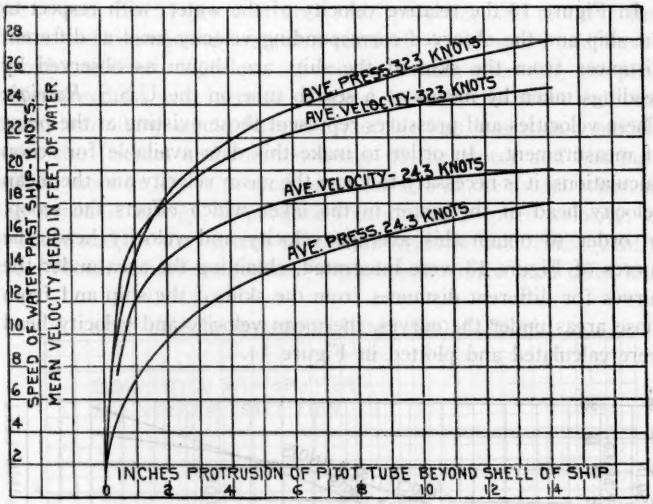

MEAN VELOCITY HEAD IN FEET OF WATER
SPEED OF WATER PAST SHIP-K NOTS.
AVE. PRESS. 32.3 KNOTS
AVE. VELOCITY-32.3 KNOTS
AVE. VELOCITY- 24.3 KNOTS
AVE. PRESS. 24.3 KNOTS
INCHES PROTRUSION OF PITOT TUBE BEYOND SHELL OF SHIP

FIG. 14.

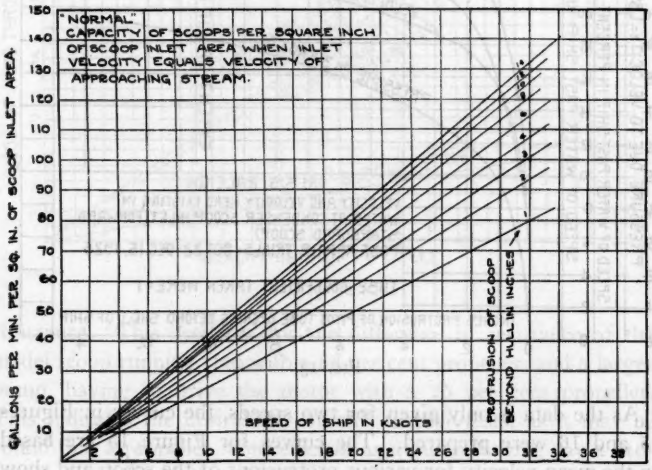

"NORMAL" CAPACITY OF SCOOPS PER SQUARE INCH OF SCOOP INLET AREA WHEN INLET VELOCITY EQUALS VELOCITY OF APPROACHING STREAM.
GALLONS PER MIN. PER SQ. IN. OF SCOOP INLET AREA
SPEED OF SHIP IN KNOTS
PROTRUSION OF SCOOP BEYOND HULL IN INCHES

FIG. 15.

stream lines affect the performance. This is given for from 1 to 14 inches of protrusion. Figure 16 gives the velocity head in feet of water for the same conditions. These curves are the same as those published in the errata sheet in the May, 1930, issue of the JOURNAL, volume XLII, No. 2, as Figures 30 and 31. From Figure 16 and Figure 10, the curves of Figure 17 were calculated, these being the predicted performance at various speeds for the one-quarter scale model scoop on the U.S.S. *Wood*. For comparison, the test results for 30 knots have been plotted, as well as the calculated curve derived from the tests with air, Figure 7 and Figure 16.

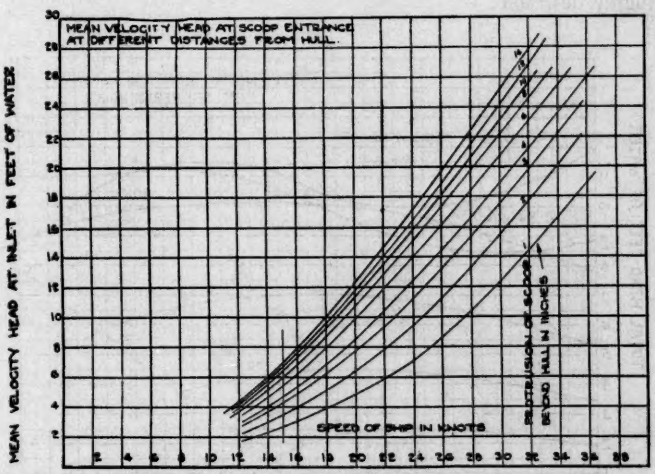

MEAN VELOCITY HEAD AT SCOOP ENTRANCE AT DIFFERENT DISTANCES FROM HULL.
MEAN VELOCITY HEAD AT INLET IN FEET OF WATER
SPEED OF SHIP IN KNOTS
PROTRUSION OF SCOOP BEYOND HULL IN INCHES

FIG. 16.

TESTS OF A ONE-QUARTER SCALE MODEL SCOOP ON THE U.S.S. "WELBORN C. WOOD."

Because of a feeling that considerable uncertainty exists in regard to the flow of water in the layers immediately surrounding the ship's skin, it was thought desirable to conduct model scoop tests on an actual ship and as a result of hearty cooperation between the United States Navy, U. S. Coast Guard and the U. S. Shipping Board, it was decided to build a ¼ scale model of a scoop

such as would be employed on a ship of the cruiser class, and test this on one of the destroyers in the U. S. Coast Guard Service. Accordingly, a model scoop, pump and overboard discharge was prepared to be installed on the U.S.S. *Wood*.

As it had been suggested that there might be some question as to the value of tests made with a venturi meter instead of a ¼ scale model of a condenser, this question was discussed at length. It was decided however to use the meter as this furnishes the only simple and reliable method of obtaining the actual volume passed and is the only method by which the complete pressure volume characteristic of the scoop could be obtained, which was of course highly desirable.

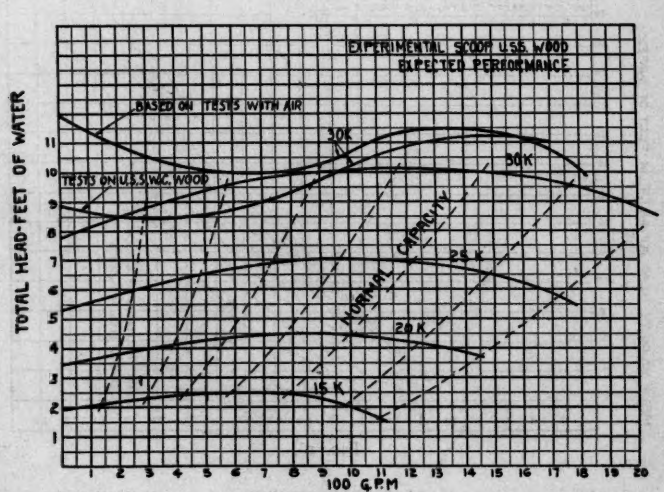


FIG. 17.

Had a model condenser and its water boxes been made, the volume passing could have been obtained by calibrating the pressure drop across the condenser and the volume thus determined. However since it would be impossible to vary the resistance through the condenser, but one point on the pressure-volume curve could have been obtained.

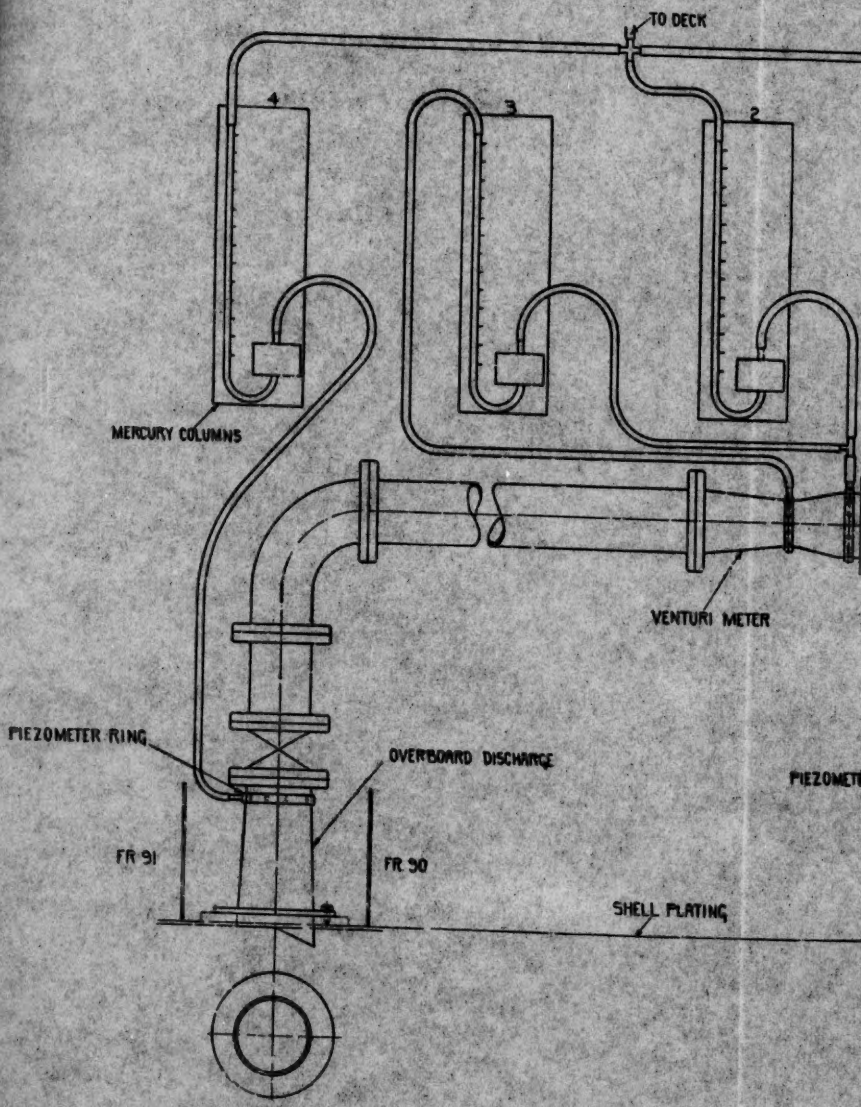
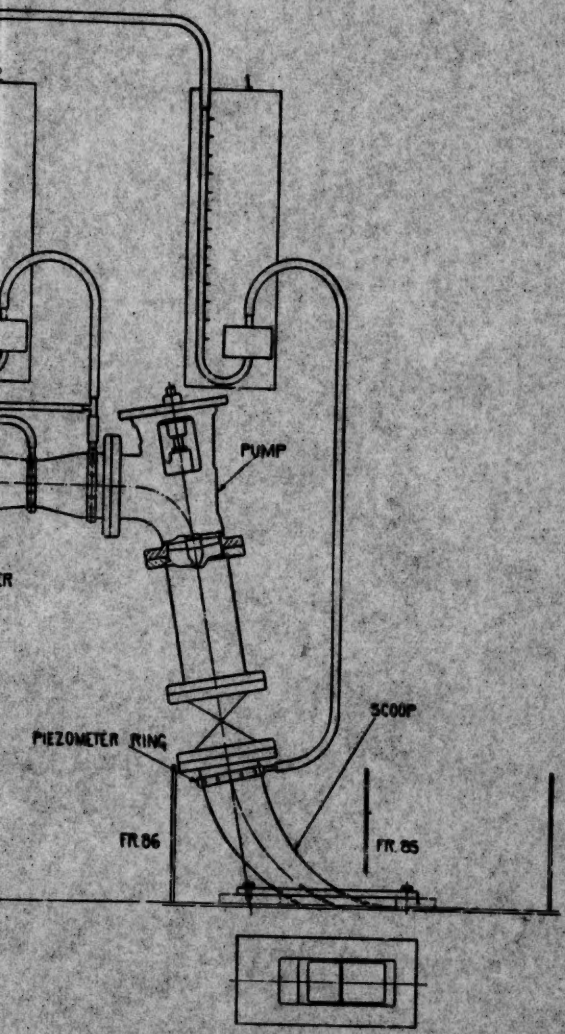

TO DECK
4
3
2
MERCURY COLUMNS
VENTURI METER
PIEZOMETER RING
OVERBOARD DISCHARGE
FR 91
FR 90
SHELL PLATING

FIG. 18.


PUMP
SCOOP
PIEZOMETER RING
FR.86
FR.85

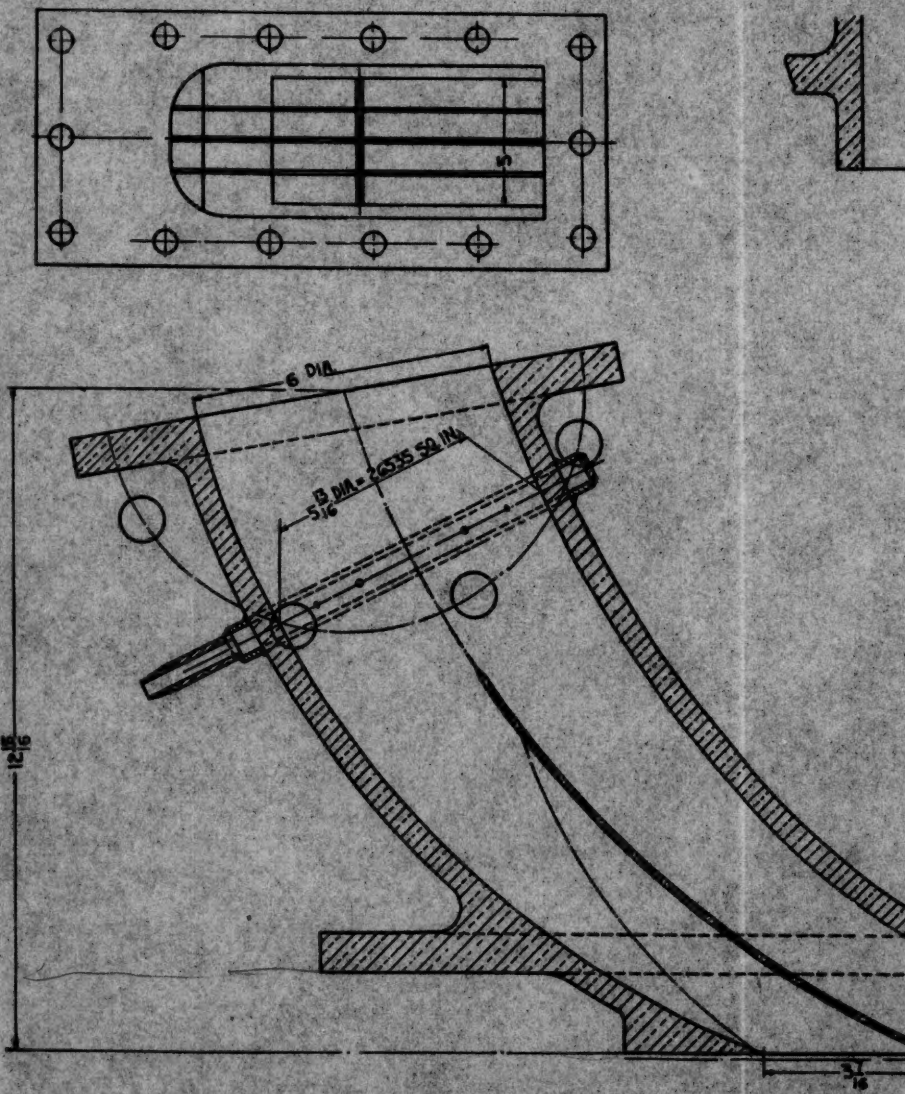


FIG. 19.

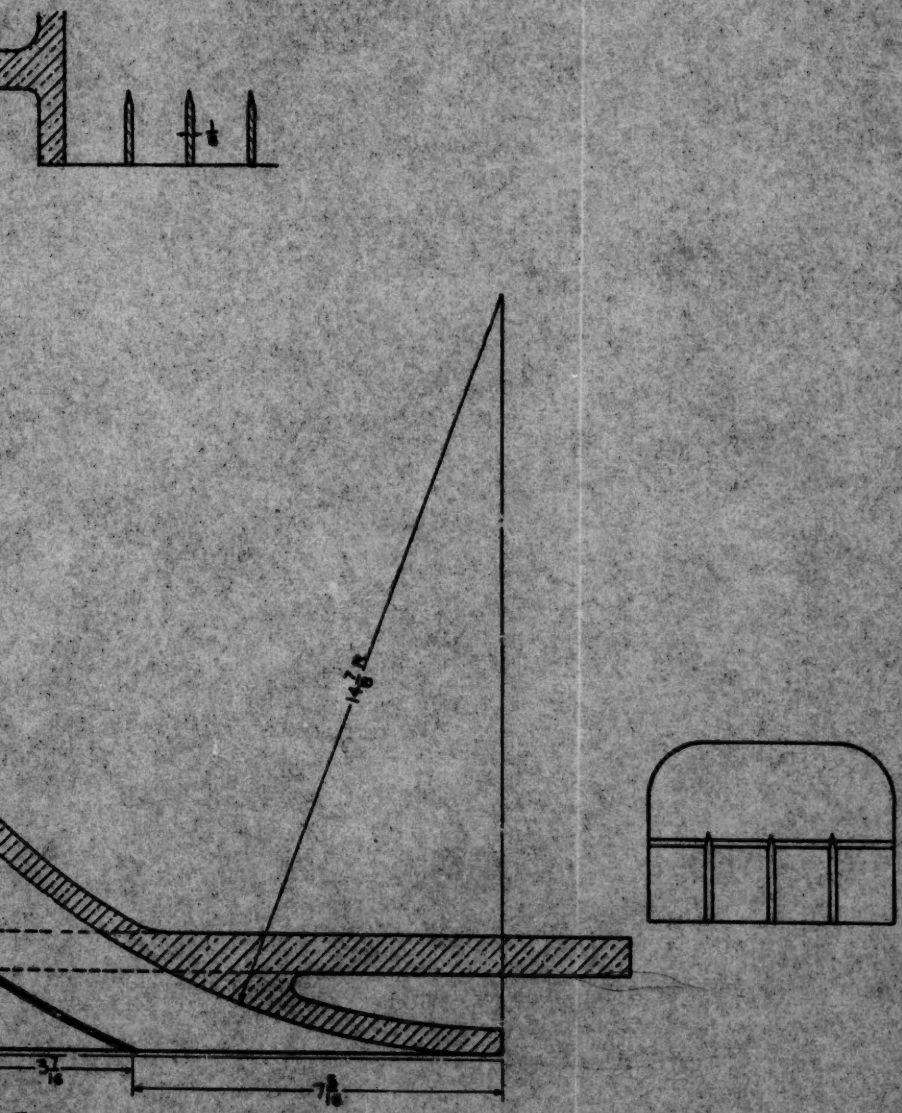


FIG. 19.

Certainly, if the scoop and overboard discharge pipe are capable of maintaining a certain difference of pressure with a given volume passing, it is entirely immaterial what causes the resistance through which the flow takes place.

A careful survey was made of the ship and the only available space which could be found was on the port side, in the after fire room, between the boilers. The scoop entrance was located between frame 85 and 86, about 3 feet outboard from the centerline of the keel and about 3 or 4 feet inboard and about 25 feet aft of the beginning of the forward docking keel.

Figure 18 shows the general test arrangement of scoop, pump, venturi meter, overboard discharge and mercury columns for recording the various pressures.

As it was necessary to use one boiler in the after fire room to obtain 30 knots, it was necessary to carry air pressure in the fire room and as this would affect the readings of the mercury columns, open ended mercury columns Nos. 1, 2 and 4 were connected to a pipe, leading above deck, so as to maintain normal atmospheric pressure on top of these gages, and therefore render them independent of fluctuations of the fire room air pressure.

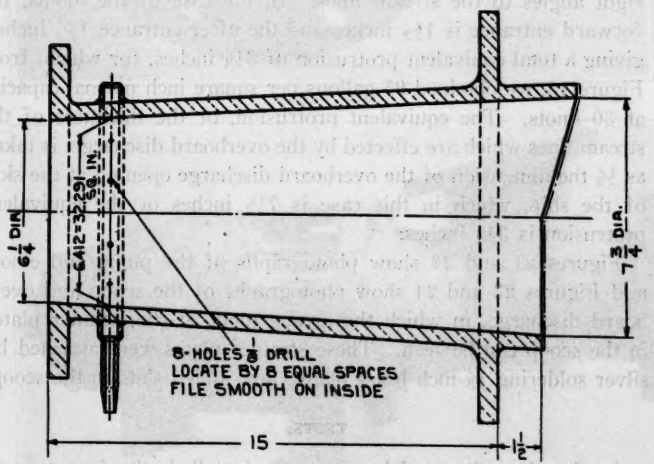


FIG. 20.

As is seen from the general arrangement, the doubling plates around the scoop and overboard discharge opening were arranged inside of the shell plating, thus leaving the scoop entrance absolutely flush with the skin of the ship and the after side of the overboard discharge opening was also flush with the hull, the lip on the forward half of the overboard discharge opening extending 2 inches beyond the hull, corresponding to a protrusion of 8 inches on the full size model. Detail drawings of the scoop and overboard discharge are shown in Figures 19 and 20 and these show the location of the piezometer rings for measuring the scoop discharge pressure and overboard discharge pressure. The drawing also gives the area at these points. Shop measurements of the entrance area of the scoop gave an entrance area of between 14.5 and 15 square inches, which, from Curve, Figure 15, establishes the normal capacity of the scoop at 30 knots between 1380 and 1425 gallons per minute or say 1400 gallons per minute. Because of the shape it is difficult to get the exact measurement of the inlet area, which of course is taken at right angles to the stream lines at the entrance. The equivalent protrusion of the lipless scoop is taken as being the same as the inlet dimension taken at right angles to the stream lines. In the case of the model, the forward entrance is 1⅝ inches and the after entrance 1½ inches, giving a total equivalent protrusion of 3⅛ inches, for which, from Figure 15, we obtained 95 gallons per square inch normal capacity at 30 knots. The equivalent protrusion, or the thickness of the stream lines which are effected by the overboard discharge, is taken as ½ the dimension of the overboard discharge opening at the skin of the ship, which in this case is 7½ inches or the equivalent protrusion is 3¾ inches.

Figures 21 and 22 show photographs of the pump and elbow and Figures 23 and 24 show photographs of the scoop and overboard discharge, in which the arrangement of the strainer plates in the scoop can be seen. These strainer plates were attached by silver soldering ⅛-inch brass plates, into milled slots in the scoop.

TESTS.

At the time the model scoop was installed, the bottom was cleaned and painted. The plating was in good condition, but as the ship is over ten years old, the shell plating is of course not as

FIG. 21.

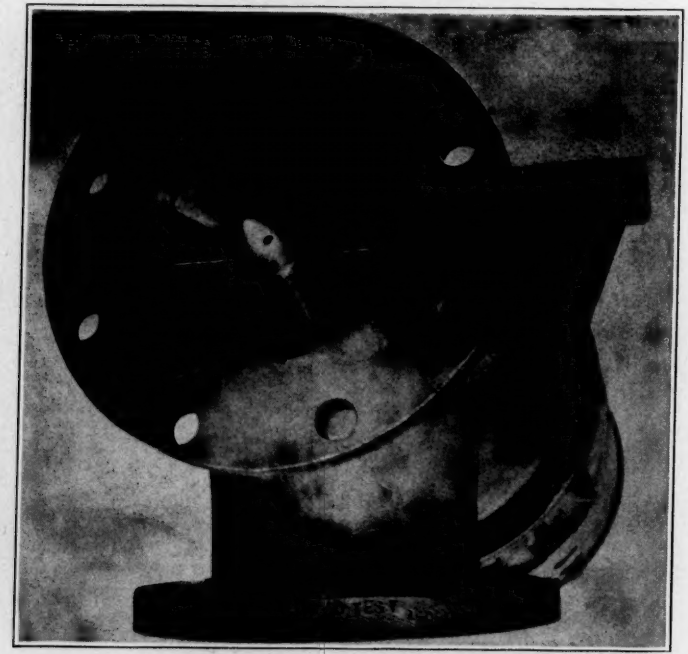


FIG. 22.

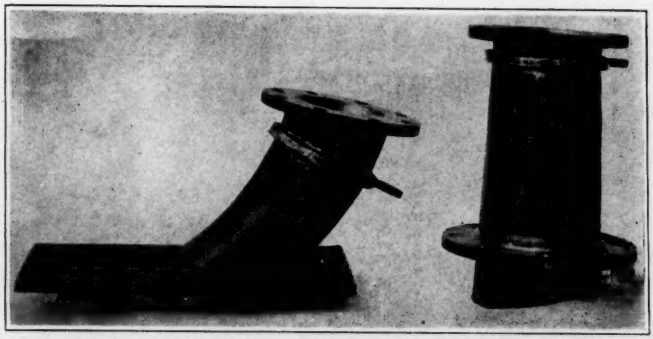


FIG. 23.

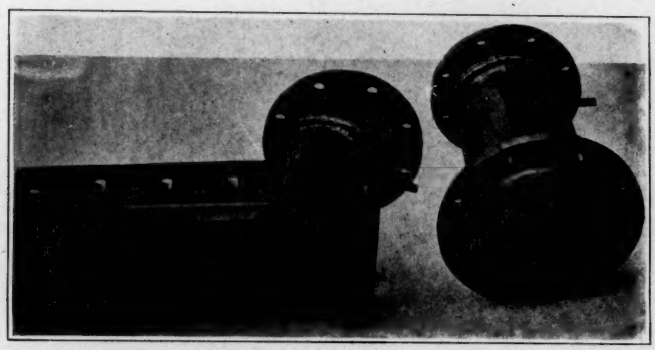


FIG. 24.

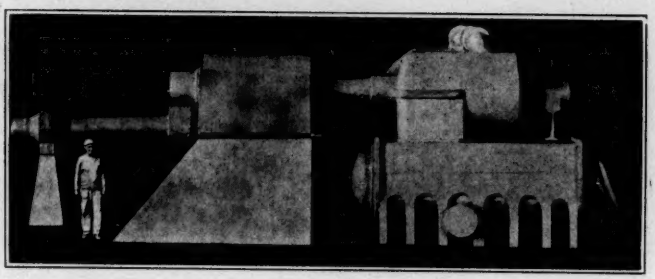


FIG. 30.

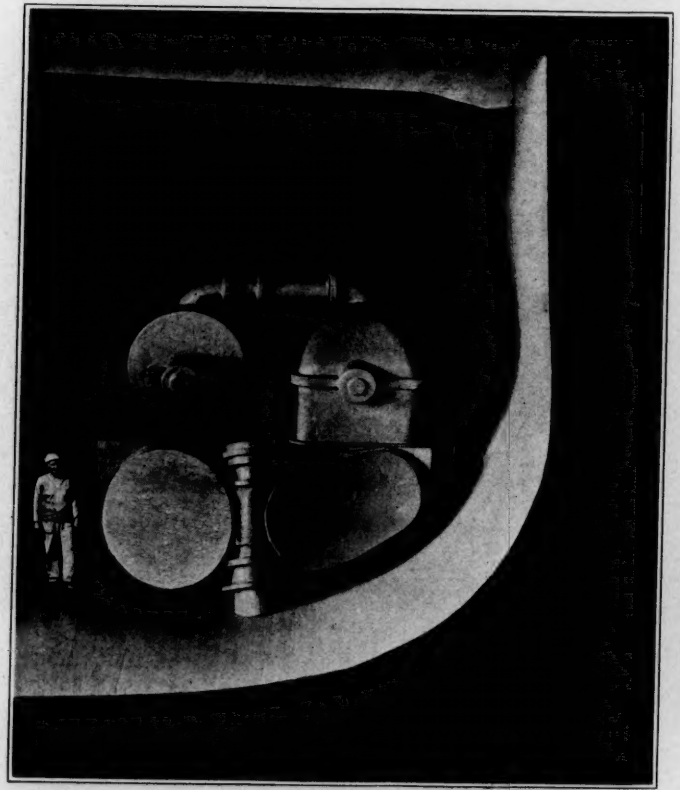


FIG. 31.

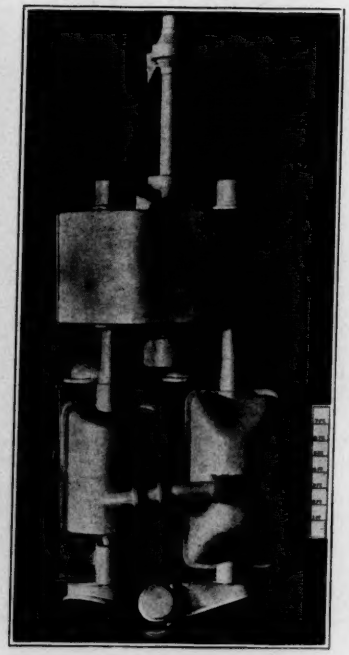


FIG. 32.

smooth as a new ship, and the surface may be considered rough considering the size of the model.

At the time of the trials about two months had elapsed since docking. It is intended to repeat the tests again just prior to the next docking to determine the effect of fouling on the performance. The results of these trials will be presented to the Society later.

After preliminary runs to check the operation of the gages, etc., official tests were run on June 11, 1931. These tests were witnessed by officers of the Navy, Coast Guard and representatives of the various shipbuilders, naval architects and engineering colleges.

These tests were run over a course outside of Fishers Island and into Block Island Sound and return. This course was selected

TABLE 1.

OBSERVED DATA								CALCULATED DATA			REMARKS
READING Nº	TIME	SHIP SPEED KNOTS	SCOOP DISCH. PRESS. GA #1 INCHES OF MERCURY	PUMP DISCH. PRESS. GA #2 INCHES OF MERCURY	VENTURI DEFLECTION GA #3 INCHES OF MERCURY	OVERBOARD DISCH. PRESS. GA #4 INCHES OF MERCURY	PUMP IDLING SPEED REVS. PER. MIN.	TOTAL PRESS AT SCOOP DISCH. FT. SALT WATER (COL. 4 + HD FROM COL. 6 + DEPRESSION	EFF. STATIC HD. BET. SCOOP & OVERBD DISCH. PRESSURE RINGS - FT. SALT WATER (COL. 4 - COL. 7)	CAPACITY OF SCOOP (FROM COL. 6)	
1	2	3	4	5	6	7	8	9	10	11	12
1	11:20	10 K	5.0	4.4	3.4	3.55		1.81	1.62	645	
2		"	5.35	5.35	.0	3.5		1.24	2.05	0	
3		"	5.3	5.2	.97	3.55		1.36	1.94	345	
4		"	5.15	4.9	2.05	3.65		1.53	1.72	500	
5	11:37	"	4.9	4.55	2.9	3.8		1.55	1.22	595	
6	11:45	20 K	7.5	7.2	1.0	.7		4.21	7.43	0	
6_a		"	7.2	7.2	1.0	.7		4.1	7.22	350	
7		"	7.0	7.0	2.0	1.5		4.20	6.1	495	
8		"	7.1	7.0	3.5	1.2		4.72	6.54	654	
9		"	7.0	6.5	4.2	1.2		4.80	6.43	717	
10		"	6.8	6.2	5.8	1.4		5.03	5.98	844	
11		"	6.0	4.6	10.0	2.0		5.27	4.44	1110	
12	12:00	"	6.5	5.4	7.6	1.7		5.17	5.32	965	
13	12:10	25 K	6.5	4.0	15.0	2.8*		7.63	6.75	1360	
14		"	7.2	5.2	12.5	2.5		7.75	7.87	1237	
15		"	7.8	6.5	10.2	2.2		7.77	8.87	1118	
16		"	8.1	7.1	8.1	1.7		7.50	9.75	995	
17		"	8.2	7.5	5.8	1.7		7.02	9.86	844	
18		"	8.2	8.0	4.1	1.9		6.57	9.65	708	
19		"	8.1	8.0	2.0	2.0		5.85	9.41	495	
20		"	8.5	8.5	.0	1.6		5.75	10.4	0	
21		"	8.4	8.4	1.0	1.9		5.89	9.87	350	
22		"	8.1	8.0	3.1	2.1		6.16	9.32	616	
23		"	7.8	7.5	5.0	2.1		6.37	8.98	784	
24		"	7.8	7.3	7.2	2.0		6.97	9.09	937	

* COL 7 - ITEM 13 OVERBOARD DISCHARGE GAUGE LOWERED 32 7/8" AT BEGINNING OF 25 KNOT RUN

STATIC READING ON GAUGES FOR SCOOP DISCH. PRESS., PUMP DISCH. PRESS. AND OVERBOARD DISCH. 19.45 in. HG.

STREAM LINE DEPRESSION, 10 KNOTS 0.191 FT.
20 KNOTS 0.766 FT.
25 KNOTS 1.200 FT.
30 KNOTS 1.720 FT.

TABLE 1. (CONT'D)

OBSERVED DATA								CALCULATED DATA			
READING Nº	TIME	SHIP SPEED KNOTS	SCOOP DISCH. PRESS. GA #1 INCHES OF MERCURY	PUMP DISCH. PRESS. GA #2 INCHES OF MERCURY	VENTURI DEFLECTION GA #3 INCHES OF MERCURY	OVERBOARD DISCH. PRESS GA #4 INCHES OF MERCURY	PUMP IDLING SPEED REVS. PER. MIN	TOTAL PRESS AT SCOOP DISCH FT. SALT WATER (COL 4 + HD. FROM COL 6 + DEPRESSION	EFF. STAT. HD. BET. SCOOP & OVERBD DISCH. PIEZOMETER RINGS-FT. SALT WATER (COL 4 - COL 7)	CAPACITY OF SCOOP (FROM COL 6)	REMARKS
1	2	3	4	5	6	7	8	9	10	11	12
25		"	7.3	6.3	9.1	2.25		6.92	8.25	1060	
26		"	7.5	5.7	11.0	2.4		7.64	8.31	1160	
27		"	7.2	4.8	13.1	2.55		7.91	7.81	1268	
28	1230	"	6.5	3.8	14.8	2.8		8.03	6.76	1350	
29	1255	30K	7.7	4.7	21.5	2.2	1500	11.21	9.65	1625	
30		"	7.5	4.7	22.0	2.1		11.14	9.55	1640	
31		"	8.8	6.5	17.0	1.4	1200	11.23	11.8	1445	
32		"	9.2	8.0	15.0	.9		11.07	12.8	1357	
33		"	10.0	9.0	10.	.5	215	10.62	14.1	1110	
34		"	9.8	9.3	5.9	.6		9.29	13.7	850	
35		"	10.	10.	2.1	.8		8.41	13.7	507	
36		"	11.	11.	.0	.1		8.97	15.6	0	
37		"	9.8	9.8	4.2	.6		8.78	13.8	717	
38		"	9.7	8.7	8.2	.4		9.78	13.9	1000	
39		"	9.5	8.0	13.	.9		10.86	13.1	1260	
40		"	9.2	6.8	16	1.2	1150	11.37	12.4	1400	
41		"	7.8	5.2	20.2	2.1	1400	10.97	9.87	1572	
42	1310	"	7.5	4.6	22.	2.1		11.14	9.55	1640	
43	1315	30K	9.	7.	17.5	1.2	1250	11.55	12.2	1465	
44	1325	25K	7.3	6.0	12.5	2.4	1000	7.82		1242	
45	1332	20K	6.8	5.6	6.4	4.1		5.18		885	
46		20K	.9	9.8	19.5	2.7	3450				
47	1345	10K	5.0	4.7	1.9	6.2		1.41		482	

* COL. 8 - ITEM 46 THIS IS PUMP OPERATING S/PEED WITH MOTOR DRIVING

because it provided the deepest water available in this locality and the best protection, as it was necessary to have relatively quiet water, in order to secure satisfactory readings of the mercury columns. The weather was very favorable, there being only a slight ground swell and a light wind.

The readings at 10 knots and 20 knots were made on the outward trip and the readings at 25 and 30 knots on the return trip. The runs at 20 and 25 knots were in water approximately 200 feet deep, but the run at 30 knots was in somewhat shallower water. There was a little rolling and a slight amount of pitching.

The data obtained, as read, is given in Table No. 1 to which has also been added the deduced readings converted from inches of mercury to feet of salt water, with an addition of velocity head

through the scoop piezometer ring, as well as what has been termed the stream line "depression." The readings as observed, would indicate considerable less head produced by the scoop than anticipated, but on the other hand the overboard discharge suction would appear to be an equal amount more than anticipated. It would appear that the reason for this is the wave system set up by the ship. For convenience an analogy may be made between the ship moving through the water and the flow through a venturi meter. Owing to the inertia effect of the water, as the ship advances the water is parted and the stream lines close to the ship conform to to the ship's form, but as we go farther out from the hull, the stream lines are less deflected and at some distance from the hull, the water is entirely undisturbed. In other words, we could substitute a wall in place of the surrounding water, so that the ship

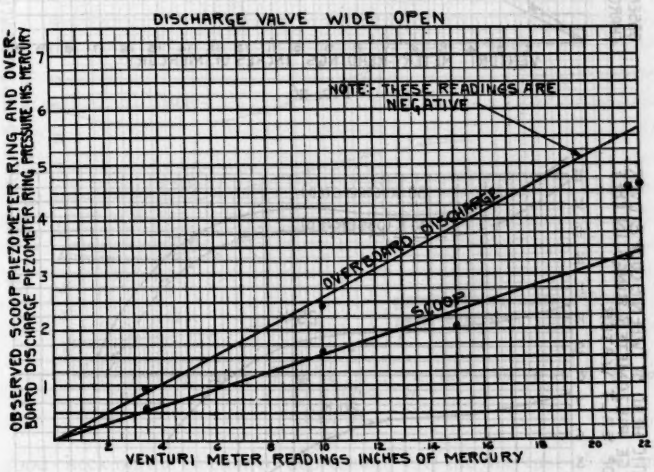

DISCHARGE VALVE WIDE OPEN
OBSERVED SCOOP PIEZOMETER RING AND OVERBOARD DISCHARGE PIEZOMETER RING PRESSURE INS. MERCURY
NOTE:- THESE READINGS ARE NEGATIVE
OVERBOARD DISCHARGE
SCOOP
VENTURI METER READINGS INCHES OF MERCURY

FIG. 25

moving through the water becomes analogous to a ship shape form inserted in a parallel pipe. The velocity of the water increases as the mid-ship section is approached, accompanied by a decrease of pressure immediately surrounding the hull and a gradual slowing up of the water along the run of the ship and a gradual increase

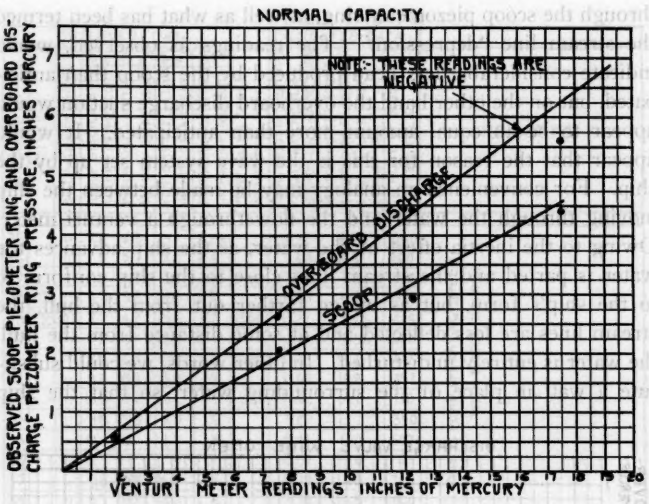

NORMAL CAPACITY
NOTE:- THESE READINGS ARE NEGATIVE
OVERBOARD DISCHARGE
SCOOP
OBSERVED SCOOP PIEZOMETER RING AND OVERBOARD DISCHARGE PIEZOMETER RING PRESSURE. INCHES MERCURY
VENTURI METER READINGS INCHES OF MERCURY

FIG. 26.

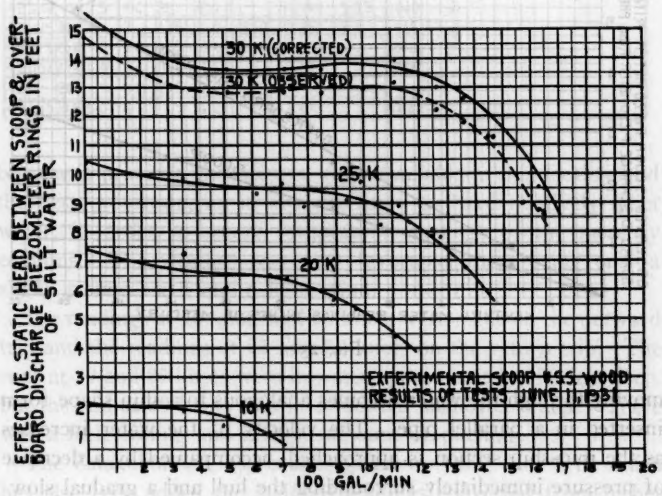

30 K (CORRECTED)
30 K (OBSERVED)
25 K
20 K
10 K
EXPERIMENTAL SCOOP 8 S.S. WOOD
RESULTS OF TESTS JUNE 11, 1931
EFFECTIVE STATIC HEAD BETWEEN SCOOP & OVERBOARD DISCHARGE PIEZOMETER RINGS IN FEET OF SALT WATER
100 GAL/MIN

FIG. 27.

of pressure back to normal. Since frames 85 and 86 are located near the mid-ship section, it is obvious that the water passing the ship must have a decreased pressure and increased velocity. As indicated in the table, this amounts to a reduction of pressure of approximately 1.72 inches at 30 knots and corresponds very closely with the depth of the observed trough opposite the location of the scoop. The fender was just 2 inches above the water line when the ship was at rest, so this furnished an easy mark from which to estimate the depth of the trough, when running at different speeds. Various observers estimated the depth of the trough at 30 knots at from 18 to 24 inches, which furnishes a reasonably

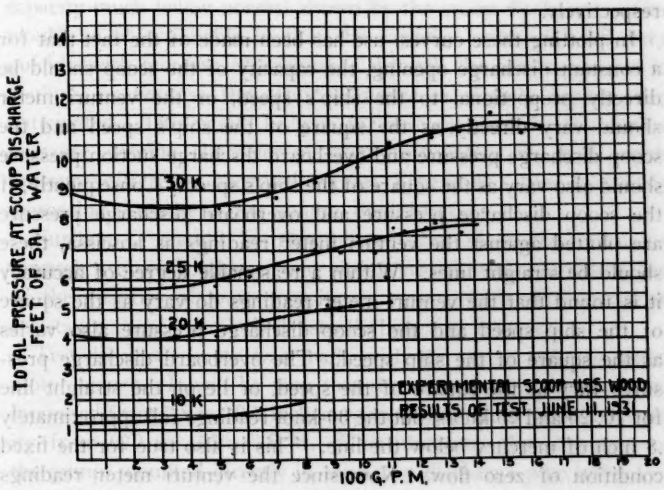


FIG. 28.

good check on the calculated depression. As the decrease of stream line pressure affects both overboard discharge and scoop equally, it is obvious that the total difference in head, between the inlet and overboard discharge, is unaffected by the estimate of this value and consequently, though there be some who will not agree with the calculations of the stream line depression, it is impossible to get away from the observed difference of pressure, as plotted in Figure 27.

In spite of the fact that the sea was relatively smooth, the motion of the ship was sufficient to cause fluctuations of the gage readings, particularly on the return trip when the sea was on the port bow. The fluctuations of the readings at 30 knots amounted to as much as 2 inches and consequently, in spite of the fact that each gage was read by a separate observer, as nearly simultaneously as possible, there are some variations and slight discrepancies in the readings. This is seen by the plotted values in Figures 27 and 28. In order to fair up the readings, the observed values of the readings at different speeds for the condition of the discharge valve wide open and the discharge valve in a fixed position, corresponding to normal capacity, have been plotted in Figures 25 and 26, respectively.

In plotting these curves, use has been made of the fact that for a constant discharge opening, the capacity of the scoop should be directly proportional to the ship's speed, or the venturi meter should vary directly as the square of the ship's speed and the scoop discharge pressure and overboard discharge suction pressure should also vary as the square of the ship's speed. Consequently, if the scoop discharge pressure and overboard discharge pressure are plotted against the venturi meter readings as abscissa, these should be straight lines. Within a reasonable degree of accuracy it is found that the venturi meter readings do vary as the square of the ship speed and the scoop discharge pressure also varies as the square of the ship speed. The overboard discharge pressures vary as the square of the speed, or lie on the straight line for 10, 20 and 25 knots, but the 30-knot readings fall approximately .8 inch of mercury below the line. This is also true for the fixed condition of zero flow. Now since the venturi meter readings and the scoop discharge readings vary as the square of the speed and for a fixed setting of the valve, the resistance or pressure drop through the system must also be proportional to the square of the speed, it is obvious that the overboard discharge readings at 30 knots must be in error by the amount of .85 inch which is the average of the discrepancy of the wide open, normal capacity, and shut-off-overboard-discharge readings. For this reason this correction has been made to the overboard discharge readings. In Figure 27, the observed readings are plotted as a dotted line and

the corrected readings as a solid line. The corrected reading varies exactly as the square of the speed, as it should, whereas the observed readings are low.

The scoop discharge pressure, corrected, is shown in Figure 28. These points as plotted, are the corrected points, as given under calculated values in the accompanying table. These readings are for various speeds for the normal capacity point and other points are observed to vary as the square of the speed, whereas without the correction previously referred to, they do not vary as the square of the speed, which would be in disagreement with the other readings.

It might at first be thought that the decrease of pressure with capacity much below normal shown by the scoop discharge curves would be objectionable in that the capacity would fall off very quickly with increase of resistance through the condenser due to fouling, or because the total resistance is slightly higher than anticipated. This would be true for a scoop alone having such a pressure volume characteristic, but fortunately the total static pressure difference obesrved, curves Figure 27, show a steadily increasing pressure as the volume diminishes.

Further, it should be remembered that since the resistance through a given condenser and piping system varies as the square of the quantity of water passing, a very considerable change in resistance causes a relatively small change in capacity.

Test 46 at 20 knots was made with the pump running under motor power. It will be seen that the venturi meter reading was increased from 7.8 to 19.5 inches, representing an increase in capacity from 970 to 1530 G.P.M. This illustrates the advantage of the pump, mounted directly in the scoop discharge pipe instead of in the bypass, as the pump can be operated in conjunction with the scoop as a booster. This is a very favorable feature for vessels which at times have to operate in southern waters, as it is possible, under these conditions to increase the circulating water from 25 to 50 per cent above that normally delivered by the scoop, at a given speed. The normal speed of this pump is 3600 R.P.M. but it would only run 3450 R.P.M. on tests, due to the fact that the wrong control was furnished and consequently it was impossible to vary the speed of the motor.

When operating in conjunction with the scoop, the speed of the motor should have been 4000 R.P.M. and the capacity would increase approximately in proportion to the revolutions. This brings out the fact that even on slow speed ships, where the speed is not high enough to justify an independent scoop installation, advantage can be taken of a scoop type sea chest which operates at any given speed to reduce the total pumping head by that amount. In the case of a 21-knot ship, the available head amounts to about 6 feet, so that with a single pass condenser, designed for a total head of 12 feet, the pumping power required is only 50 per cent of that which would be necessary for a ship fitted with an ordinary sea chest.

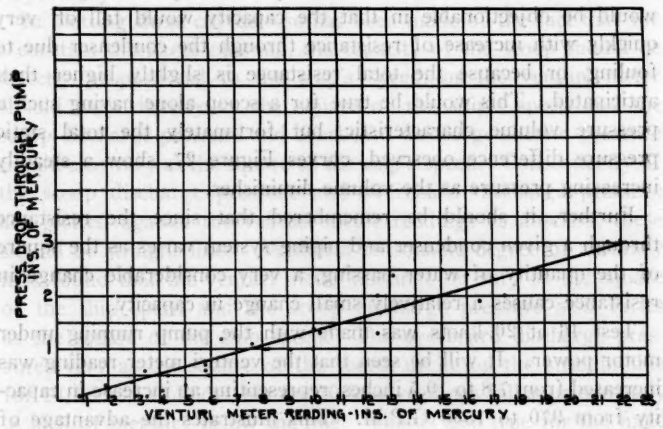


FIG. 29.

Figure 29 gives the pressure drop across the pump in inches of mercury, plotted against the venturi meter reading. The average of the points plot well on the straight line, indicating that the drop of pressure through the pump varies approximately, as the square of the quantity of water, passing through it, as it should.

It will be noted that the scoop employed in these tests had three strainer plates whereas with the lipless type of scoop, one strainer plate, in addition to the splitter, is sufficient. It should also be noted that the experimental scoops tested in the laboratory had

one splitter and no strainer plates. Consequently, as these are inserted at the point of the maximum velocity, a performance at least 10 per cent lower than the experimental laboratory scoops would have been expected. In view of this consideration, the results of the tests on the actual ship were considerably more than satisfactory and exceeded the estimated performance. From Figure 28 it is seen that at the lower capacities, the pressure produced by the scoop is proportionately lower than in the experimental scoop. This was to have been expected, since at low capacities, only the water closest to the skin of the ship enters the scoop and consequently, as the quantity of water entering the scoop decreases, the mean effective available velocity head decreases. In a full size scoop, this falling off of pressure at lower capacities would not exist, since the water increases in velocity very rapidly as the distance from the skin increases, and of course the effect of this layer is inversely proportional to the size of the scoop.

Reference to Figures 15 and 16 show this effect very clearly, since after the first two or three inches outward from the skin, the change of velocity head and velocity are much less rapid. A straight divergent tube is the most efficient means of converting velocity head into static pressure.

In all the designs of lipless scoops of which tests are here given, the area increases along a curved path from inlet to exit which is not conducive to high velocity conversion efficiency. However if we wish to turn through a given angle in a limited distance, experiments have shown that a higher efficiency is obtained by making a large radius diffusion tube, than by making a short radius turn at high velocity and then performing the velocity-pressure conversion in a straight tube.

It is obvious that the smaller the angle through which the scoop has to turn and the greater the length available for the diffusion tube the higher the efficiency of the scoop would be. If instead of an 80-degree turn and a short diffusion length, a 10-degree turn and greater length were permissible, an efficiency of 86 to 90 per cent would be possible instead of 65 per cent obtained in the scoops here shown.

In high speed Naval vessels it is seen that even with an efficiency of 65 per cent a greater head is available than is required for a

single pass condenser, so the problem becomes one of designing the lightest, most compact scoop which will give the required head. This is what has been done in the model scoop under discussion. The present scoop occupies little more fore and aft length than a straight vertical pipe.

Figures 30, 31 and 32 are photographs of a model of the underneath condenser with vertical overboard discharge pipes and a lipless scoop combined with a new type water box, in which the propeller pump is built integral with the water box. As the pump is located between the two radial flow tube banks, the water box is fitted with two covers, permitting the removal of all the tubes in the condenser without dismantling any piping, dismantling the

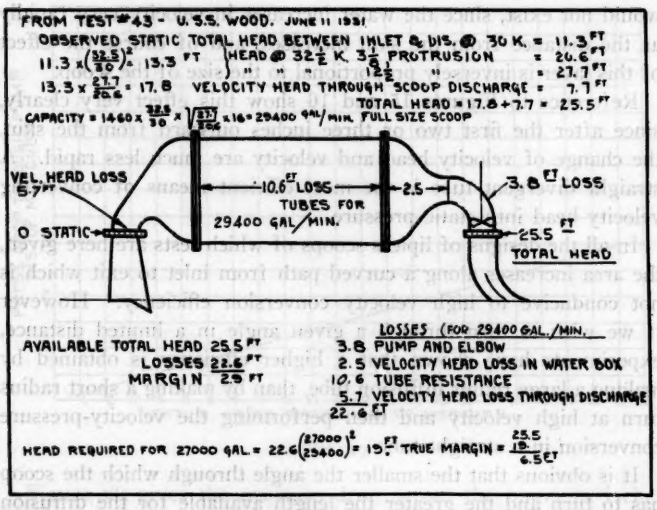

FROM TEST #43 - U.S.S. WOOD. JUNE 11 1931
OBSERVED STATIC TOTAL HEAD BETWEEN INLET & DIS. @ 30 K. = 11.3 FT
11.3 x (32.5/30)² = 13.3 FT HEAD @ 32½ K. 3½ PROTRUSION = 20.6 FT
" " " 12½ " = 27.7 FT
13.3 x 27.7/20.6 = 17.8 VELOCITY HEAD THROUGH SCOOP DISCHARGE = 7.7 FT
TOTAL HEAD = 17.8+7.7 = 25.5 FT
CAPACITY = 1460 x 32.5/30 x √(27.7/20.6) x 16 = 29400 GAL/MIN. FULL SIZE SCOOP
VEL. HEAD LOSS 5.7 FT
10.6 FT LOSS
TUBES FOR 29400 GAL/MIN.
2.5
3.8 FT LOSS
0 STATIC
25.5 FT TOTAL HEAD
AVAILABLE TOTAL HEAD 25.5 FT
LOSSES 22.6 FT
MARGIN 2.9 FT
LOSSES (FOR 29400 GAL./MIN.
3.8 PUMP AND ELBOW
2.5 VELOCITY HEAD LOSS IN WATER BOX
10.6 TUBE RESISTANCE
5.7 VELOCITY HEAD LOSS THROUGH DISCHARGE
22.6 FT
HEAD REQUIRED FOR 27000 GAL = 22.6 (27000/29400)² = 19. FT TRUE MARGIN = 25.5/19 = 6.5 FT

FIG. 33.

pump or removing water boxes. This makes the most compact arrangement of condenser and circulating water system which has ever been produced and gives a degree of accessibility to the tubes which has never before been obtained in any type of marine condenser. This type of water box would be impossible without the curved lipless scoop.

Attention is called to this new water box arrangement, because it is obvious, from an examination of the photograph, that the losses through the pump and water boxes in this construction will be considerably lower than those observed in the test just noted, since the elbow and connections are entirely eliminated.

Figure 33 shows a calculation of the performance of a condenser of 10,000 square feet surface, with tubes 12 feet long, designed for a circulating water capacity of 27,000 G.P.M. at 32½ knots.

Data for these calculations was taken from the observed static total head between inlet and discharge at 30 knots, for test 43 of the U.S.S. *Wood*. The full size scoop would have an equivalent protrusion of 12½ inches and as the complete calculations are given, this will be quite obvious. It should be noted that while the estimated condenser resistance at 27,000 G.P.M. is only 19 feet, the scoop as designed will have a margin of head available of 6½ feet or a capacity of approximately 30,000 G.P.M., instead of 27,000 G.P.M., which is the estimate based on calculations, made at a time when no work of any sort had been performed, either in a laboratory or on the *Wood*. An interesting point to note is that, whereas the inlet area of the full size scoop is 230 square inches, the inlet area of the standard scoop for a similar installation was 620 square inches, and whereas the capacity of the standard scoop was approximately 13,000 G.P.M. against an observed total head of about 9 feet, the present scoop, with only a little over ⅓ the area, is capable of delivering 2½ times the water, at three times the total available head.

In conclusion it may be said that the tests indicate that there seems to be little ground for the belief frequently expressed that the poor performance of scoops is due to a very thick layer of eddying water adjacent to the skin of the ship, as appears to be true if one could judge by the eddying mass of water at the surface, a foot or more wide which is seen when looking over the side of the ship. The surface condition which is observed is due to the transition from a light to a heavy medium, and this state of turbulence disappears entirely two or three feet under the surface, depending upon the size of the ship and the distance from the bow.

Because of the close agreement between the predicted performance and the tests on an actual ship, this also confirms the practical value of the Pitot tube readings of velocity head past the ship as determined on the U.S.S. *Raleigh*, which with the laboratory tests of the scoops was used for predicting the results on the U.S.S. *Wood*.

METHOD OF CALCULATING RESULTS OF EXPERIMENTAL SCOOP TESTS ON U.S.S. "WOOD."

1. Determine volume from venturi meter equation

$$\text{G.P.M.} = 350\sqrt{\text{Venturi meter reading}}$$

2. Calculate velocity heads at scoop piezometer ring, overboard discharge piezometer ring and at overboard discharge outlet.

3. Obtain velocity head of water past ship at 3⅛-inch protrusion for scoop and 3¾-inch protrusion for overboard discharge. (From Curve Figure 16 at proper speed.)

4. From ratio $\frac{\text{Velocity head through overboard discharge}}{\text{velocity head past ship}}$ and curve Figure 41. (Scoops Vol. XLII A. S. N. E.) February, 1930. Calculate suction head of overboard discharge.

5. Add difference in velocity head at overboard piezometer and at overboard discharge outlet × 90 per cent to overboard suction head and subtract the sum from the corrected overboard discharge pressure gage reading. The remainder represents the decrease in the stream line pressure adjacent to the scoop and overboard discharge as a first approximation. (Note 90 per cent is assumed diffuser efficiency.)

6. Add the reduction in stream line pressure to the velocity head past the ship at 3¾-inch protrusion and use this new increased velocity head for recalculating the discharge suction from Curve Figure 41. This second approximation will be close enough. Repeat 5 and thus obtain the corrected reduction in stream line pressure.

7. To the scoop discharge piezometer gage reading corrected for zero reading and reduced to feet of salt water add velocity head at scoop piezomoter ring and to this add the second approximation

of the reduction of stream line pressure. This is the total head produced by the scoop. The stream line depression must be added since the pressure at the scoop inlet is lower when the ship is in motion than when at rest when the zero readings are taken.

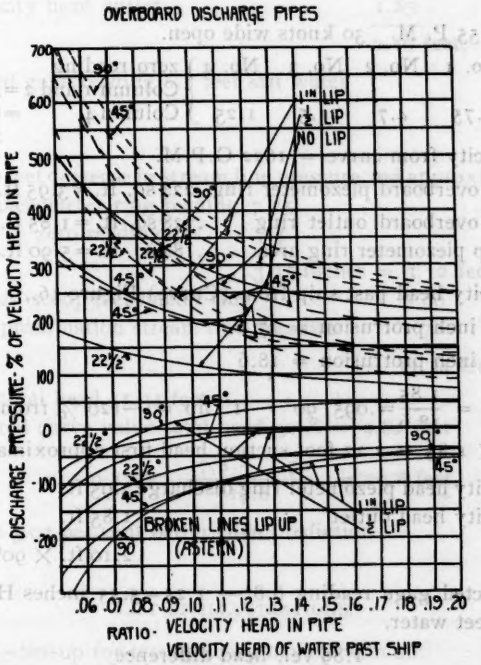


FIGURE 41. FROM FEBRUARY, 1930, JOURNAL A. S. N. E.

8. To compare scoop discharge total head with predicted curves increase the predicted curve value in the ratio of the curve value of the velocity head of water past ship at 3⅛-inch protrusion plus stream line pressure reduction to the curve value of velocity head past ship. The curve value should be that at 3⅛-inch protrusion.

NOTE: Owing to the gate valve ahead of the divergent overboard discharge pipe the velocity conversion can be figured only for the condition of wide open discharge. This value of the velocity con-

version must be used at each speed for calculating the stream line reduction of pressure and the reduction of pressure thus obtained must be used for all calculations at a given speed.

SCOOP CALCULATION EXAMPLE.

Test 12.55 P. M. 30 knots wide open.

Gage No. 1	No. 2	No. 3	No. 4	zero reading
				Column 1 and 2 = 4.4 inches
7.75	4.7	21.5	1.25	Column 4 = 6.8 inches

1. Capacity from curve = 1622 G.P.M.
2. Area overboard piezometer ring .224 sq. ft. = 3.95 ft. vel. head
   Area overboard outlet ring .328 sq. ft. = 1.85 ft. vel. head
   Scoop piezometer ring area .184 sq. ft. = 5.90 ft. vel. head
3. Velocity head past ship from (Curve) Figure 16.,
   3-1/8 inch protrusion = 17.8
   3-3/4 inch protrusion = 18.6
4. Ratio = $\frac{1.85}{18.6}$ = .095 90° — 1″ lip = —120 % from Fig. 41
   1.2 × 1.85 = 2.22 foot suction head first approximation.
5. Velocity head piezometer ring discharge 3.95 ft.
   Velocity head outlet $\frac{1.85 \text{ ft.}}{2.10 \text{ ft.} \times 90\% = 1.89}$
   Corrected gage reading 6.8 — 1.25 = 5.55 inches Hg. = 6.1 feet water.

   1.89 vel. head difference
   2.22 suction head
   4.11

   Corrected gage reading 6.10
   4.11
   1.99 decrease in stream line pressure 1st approximation.
6. 1.99 plus 18.6 = 20.59

   $\frac{1.85}{20.59}$ = .0899 gives — 135 per cent

1.85 × 1.35 = 2.49 corrected suction head 2nd approximation.

Repeating fifth with new values—

Velocity head piezometer ring discharge 3.95
Velocity head outlet 1.85
2.10 feet.

Corrected gage reading 6.1 feet salt water

1.89	6.10
2.49	4.38
4.38	1.72 feet decrease in stream line pressure 2nd approximation.

7. Scoop piezometer ring reading 7.75
zero reading 4.4
3.35 inches = 3.72 feet
vel. head scoop piezometer ring 5.90
2nd approximation stream line pressure 1.72
11.34 feet

8. Scoop total head 11.34 feet
Predicted curve value total head 30 K = 9.82 feet

$$9.82 \times \frac{17.8 + 1.72}{17.8} = 10.8 \text{ feet}$$

Test is 3.03 per cent better than prediction.

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Fig. 31.—Model, showing underneath condenser arrangement.

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THE RESISTANCE OF COPPER—NICKEL STEELS TO SEA ACTION.

Engineering, June 12, 1931.

U. S. S. "AKRON".

Navy Press.

X-RAYS IN THE STEEL INDUSTRY.*

By H. H. Lester, Research Physicist, Watertown Arsenal.

The present century now only thirty years old has witnessed phenomenal changes in science. The Electron Theory, The Quantum Hypothesis, The Bohr-Sommerfield atom, modern developments of radiation theory have revolutionized our concepts of the physical universe. The rapidity with which our scientific knowledge is growing is marvelous. Great advances have taken place in industry during this period. Buildings, bridges, machines, plants, have undergone tremendous changes. The tendency is toward greater size, greater complexity, greater power, greater output. The rapidity with which these developments have occurred is astounding. The vastness of the changes, the greatness of their effects on economic life can hardly be visualized. Fifteen years ago 300 pounds per square inch was considered high steam pressure. Today 1200 pounds per square inch is common and 3000-pound pressures are being tried experimentally. Fifteen years ago 50,000 Kw. generators were dreamed of, but none had been made. Today there is in construction a turbine that will take steam at 1200 pounds per square inch pressure, and discharge it into a vacuum and that will drive a generator delivering over 200,000 Kw. The magnitude of this industrial machine may be realized better when we consider that the average load in Metropolitan Boston today is around 200,000 Kw. This single machine will be capable of delivering electrical energy to supply the needs of a city of 1,000,000 people. Illustrations of colossal growth could be selected from railway transportation, from chemical manufacturing, from oil refining, from most of the fields of economic production.

It is not accidental that parallel development occurs in science and industry. The two fields of human endeavor are related. Each responds to stimulus from the other. Probably neither could develop without the other. Should progress in the development of the machines of industry cease, stagnation in the progress of scientific thought would follow, and the evolution of civilization would be interrupted. It is vital to society that this progress should not stop and that industry continue its upward and onward trend.

The engineer is the builder for industry. From a sure base of definite achievement he erects in his mental vision greater edifices, more powerful mechanisms. The concrete realization of his dreams requires building materials, the more important of which are metals, chiefly steels. For these he turns to the metallurgist. Consequent to the rapid development of modern industry there has been a rapid development of alloy steels. There has been, and is, an ever increasing demand for steels of higher and higher physical properties, and for steels of specialized properties. This demand has been met by studies to improve processes and to improve chemical compositions. Much of the investigative work has been purely empirical, partly because of the fact that in spite of the millions of tons of steel in daily use, its fundamental nature has not been well understood. In fact certain microstructures in steel are today matters of controversial argument. This lack of understanding of the nature of steel is no reflection on the metallurgists who have devoted years to its study, it has been due for the most part to the absence of adequate methods of study. Steel is very heterogeneous and highly complex. It owes its intrinsic physical properties to microscopic and to submicroscopic structures within the metal.

There has been developed in recent years new tools of research that will permit the study of these fine structures. The microscope in the hands of F. F. Lucas has reached hitherto unexplored regions in the field of microscopy. Even his remarkable achievements, however, have failed to

* Released for Publication by Authority of the Ordnance Department, U. S. Army.

picture the finest structures in steel. There has been needed an instrument of even greater resolving power. Such an instrument is found in the X-ray diffraction apparatus. This tool makes possible the charting of relative spatial arrangements among atoms themselves. X-ray studies of metals have been so fruitful during the past few years that the American Society for Testing Materials has recognized a new branch of metallurgical science, X-ray Metallography.

The empirical results of applied research are reflected quickly in modified manufacturing technique, and in the development of new products. Such research serves industry directly. The more fundamental theoretical research may not serve directly in the solution of production problems. It aims to increase the sum of knowledge regarding basic facts and principles. It serves pure science. The X-ray diffraction method for the study of metals comes within the category of fundamental rather than applied research. Its more important achievements do not affect production directly, but do supply much needed basic information that will guide and direct applied research. To illustrate, X-ray diffraction studies do not give specific information about the hardening of copper, but they do give very definite information as to the nature of hardness. With this information, it is possible and probable that copper will be hardened as effectively as aluminum has been. The actual hardening of copper will be a problem in applied research.

In the hands of such men as Hull, Davey, Bain, Jefferies, Archer, McKeehan, Clark, and a host of others in this country and abroad, this new tool has already produced remarkable results in clearing up the mystery that has enshrouded steel. We know now the nature of hardness. We know why steels are strong or weak. We have a clear picture of a solid solution. We know how metals flow in plastic deformation. A few tenths of one per cent of carbon added to iron changes iron, a relatively weak substance to steel, our strongest building material. This fact has been known for centuries. We can state pretty definitely now the exact rôle of the carbon atom in this transformation, and the part it plays in the quenching and tempering of steel.

Such a powerful tool deserves description. Figure 1 shows one arrangement of X-ray tube, metal specimen, and photographic film. The X-ray beam is passed, in this case, across the surface of the metal. The rays are diffracted from many small crystals in the metal and diffracted beams from different atomic planes in the crystals are registered at different points on the photographic film. Figure 2 is a drawing taken directly from developed photographic films, showing typical X-ray crystal spectra from two steels. The diagram shows the arrangement of the iron atoms in the crystal patterns. This pattern was worked out from measurements of the relative positions of the spectrum line. The P_2J was found to have a slightly enlarged crystal unit, due presumably to the presence of an alloying constituent that was held in solid solution in the iron. Figure 3 illustrates spectra from three different steels. One is a plain carbon steel, one a 13 per cent manganese steel, and the other a tool steel. The first line marks the position of the undeviated beam, the other lines represent images from the rays diffracted by different atomic planes. Measurements of these positions relative to the undeviated line give data that permit the construction of the crystal patterns. Some idea of the magnitudes involved may be gained from a study of the table in the figure. With a good film different readings of the line positions can be duplicated within about ± .010 inch, for a line corresponding to that marked D in the figure. The error in measurement is approximately ± .1 per cent. The 10.989 inches on the film converted to interatomic spacings within the crystal means .00000000221 inch or .561 angstrom unit (1 angstrom $= 1 \times 10^{-8}$ cm.). This measurement is certain to ± .1 per

cent that is to ± .005 angstrom unit. An iron atom has a diameter of 2.48 Å. This particular measurement is in error by approximately ± 1/500 of an atom diameter. It is interesting to compare these magnitudes with those obtained by microscopic examination. F. F. Lucas obtains useful magnifications up to 4500 diameters. That is, he can distinguish distances on the ground glass of the camera corresponding approximately to 1×10^{-5} inches in the metal itself. Under favorable conditions, as illustrated in Figure 3, the X-ray diffraction method will distinguish distances as small as 5×10^{-11} inches. That is, the X-ray in this sense is 100,000 times more sensitive than the microscope. Such a tool is worthy of respect.

In the above it has been attempted merely to illustrate some of the possibilities of this method of study. A complete discussion could not be attempted within the limits of this paper. The Hull-Debeye type of diffraction spectra shown in Figures 2 and 3 is only one of the kinds of diffraction

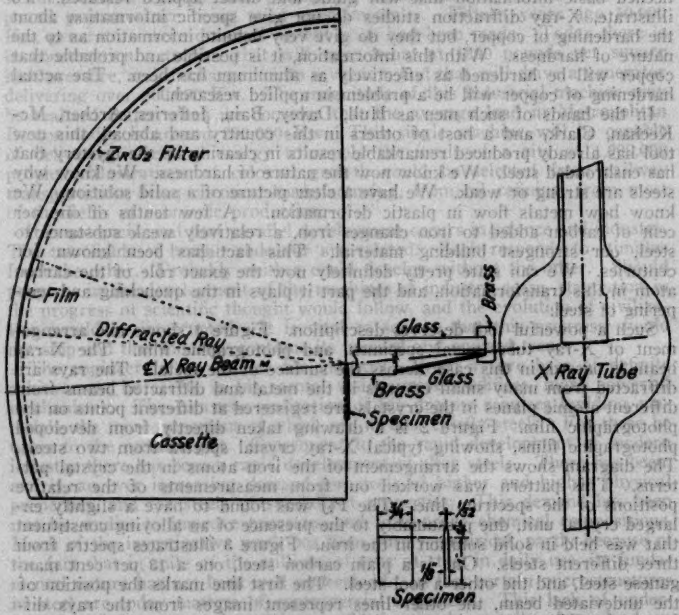


FIG. 1.—HULL DIFFRACTION TUBE SLIT SYSTEM, AND DIFFRACTION CAMERA.

X-rays proceed from the anode of the X-ray tube. After passing through a slit system that constructs the rays into a flat beam, they pass over the surface of a metal specimen. The minute crystals in the surface reflect the part of the beam at different angles according to the different plane systems in the crystal. The reflected rays pass through a filter and are caught on a photographic film that is bent around a cylindrical surface that has the surface of the metal specimen in its axis.

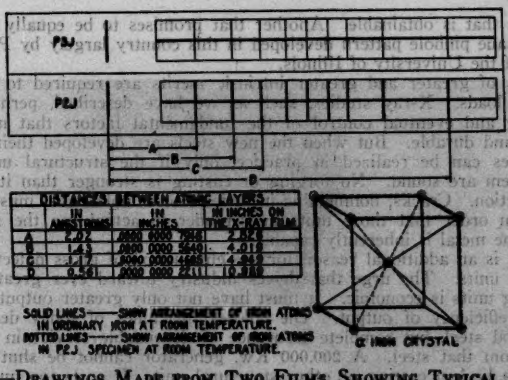


FIG. 2.—DRAWINGS MADE FROM TWO FILMS SHOWING TYPICAL X-RAY CRYSTAL SPECTRA.

The heavy line to the left is the image of the undeviated beam. The lighter lines represent reflections from different atomic plane systems in the crystal. The table of measurements for four of the lines gives the relationship between the film magnitudes and the corresponding magnitudes within the metallic crystals. This figure illustrates also a practical application of diffraction data. The P_8J specimen showed peculiar physical properties. This was traced to a different sized crystal pattern presumably due to the fact that the P_8J specimen contained an impurity that was in solution in the iron.

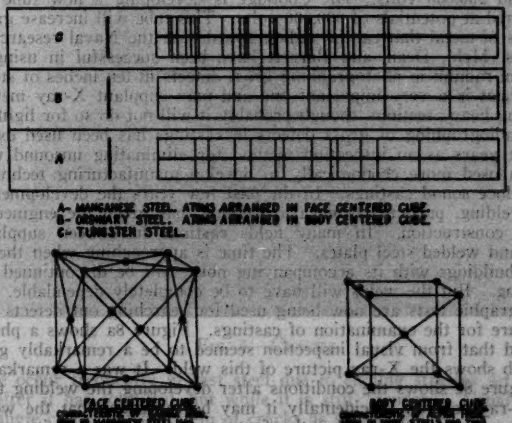


FIG. 3.—DRAWINGS FROM DIFFRACTION FILMS.

Crystal spectra from three different steels. The atomic arrangements in the crystal patterns worked out from films *A* and *B* are shown drawn to scale.

patterns that is obtainable. Another that promises to be equally important is the Laue pinhole pattern developed in this country largely by Prof. G. L. Clark of the University of Illinois.

Steels of greater and greater intrinsic merits are required to carry the modern loads. X-ray studies, such as we have described, permit understanding and eventual control of the fundamental factors that make steel strong and durable. But when the new steels are developed their superior properties can be realized in practice, only if the structural units made from them are sound. No forging or casting is stronger than its weakest vital section. Cracks, nonmetallic inclusions, blow holes, etc., must be eliminated in order that those units may render in actual use the service of which the metal is inherently capable.

There is an additional reason for the elimination of gross defects in steel building units. The urge that drives industry toward ever greater manufacturing units is economic. It must have not only greater output, but also greater efficiency of output. The colossal machines of today demand not only good steel, but complete assurance of dependable service in the units made from that steel. A 200,000 Kw. generator cannot be shut down to replace a broken casting or other structural unit. It costs too much to interrupt that amount of production even for a moment. The casting or forging *Must Not Fail.*

Here again the X-ray is of service. Castings and welded structures have in the past been peculiarly subject to gross defects. These defects hidden within the metal may be photographed by means of X-rays, in much the same way that the physician photographs a broken bone, or the dentist X-rays defective teeth. Figure 4 illustrates how such pictures may be obtained. Figures 5, 6, 7 illustrate typical X-ray negatives of casting defects. At present commercially available apparatus can be used for practicable application of the method to steel sections up to 3 or 3½ inches in thickness. To get pictures through this thickness requires potentials on the X-ray tube of around 250,000 volts. Dr. Coolidge is developing a new tube that can be operated at potentials of 900,000 volts. This tube will increase greatly the thickness of metal that can be radiographed. At the Naval research laboratory Drs. Mehl, Dean, and Barrett have been successful in using gamma rays from radium in photographing gross defects in ten inches of steel. This development is a very important one and may supplant X-ray methods for examining heavy sections, though probably it will not do so for light ones.

The radiographing of gross defects in castings has been used extensively in recent years as an inspection method for eliminating unsound units. It has been used more economically to develop manufacturing technique that will produce sound castings. In the past few years the development of the art of welding, particularly electric welding, has given the engineer a new form of construction. In many fields castings have been supplanted by formed and welded steel plates. The time is approaching when the riveting of steel buildings with its accompanying noise will be discontinued in favor of welding. But the welds will have to be completely dependable.

Radiographic tests are now being used for searching out defects in welds as they are for the examination of castings. Figure 8a shows a photograph of a weld that from visual inspection seemed to be a remarkably good one. Figure 8b shows the X-ray picture of this weld. It was a remarkably poor one. Figure 8c shows the conditions after developing the welding technique under X-ray control. Incidentally it may be remarked that the weld from which 8c was taken resembled the first one so closely that it would have been difficult to distinguish them by visual examination.

This application also is used most economically to develop technique. It is better to make good welds than to reject bad ones. The X-ray control

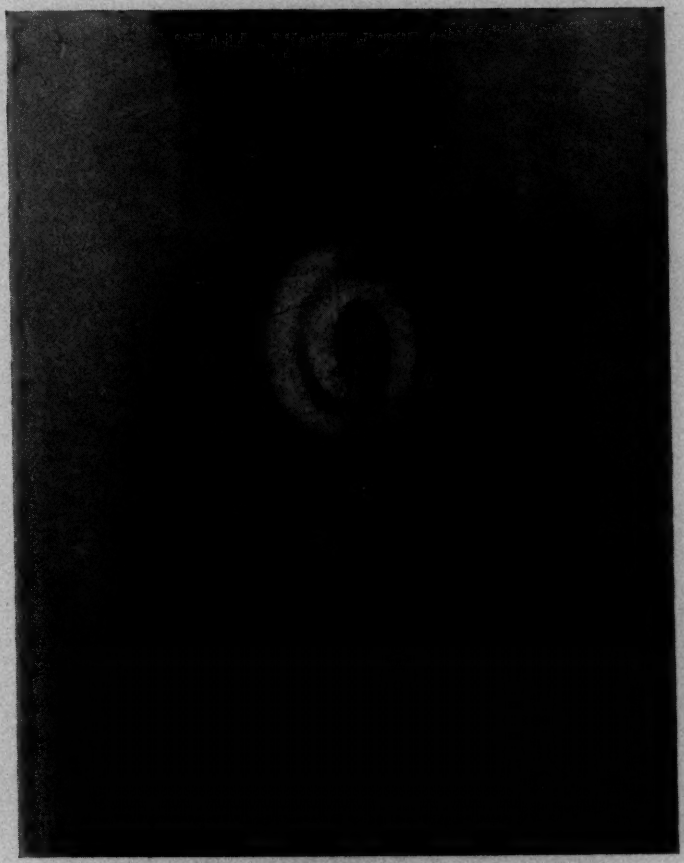


FIG. 5.—CRACKS.

These occur frequently in cast steel due to cooling stresses set up in the metal after it has solidified, but while it is still "tender." Usually there is not a single crack, but a system of cracks as shown above.

FIG. 6.—SPONGY METAL.

Metal occupies more volume in the liquid state than in the solid. When it solidifies a definite cavity may be formed (see Figure 7), or there may be a region that contains many minute cavities as in the case illustrated here.

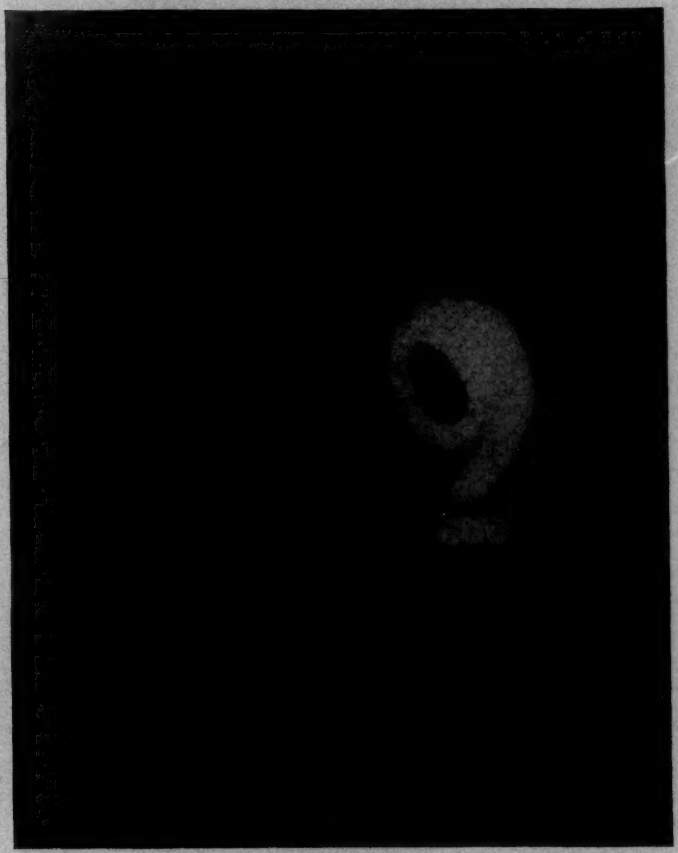
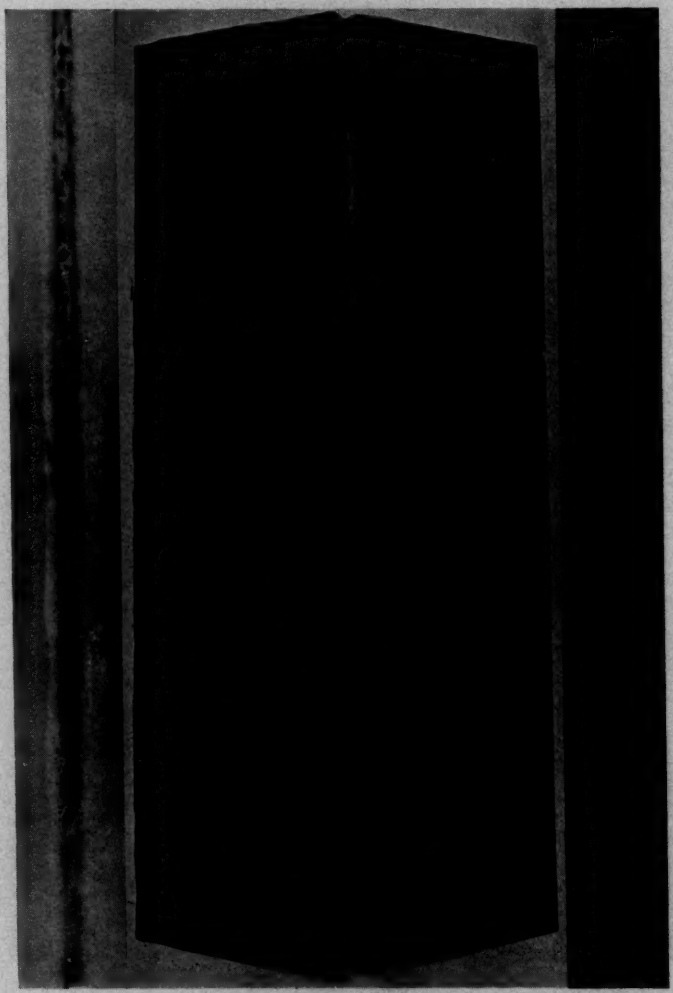


FIG. 7.—PIPE.

Molten metal is fed into the casting while it solidifies. Sometimes feeding is imperfect and pipes occur, either as definite cavities as shown above or as spongy regions illustrated in Figure 6.



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(*Army Ordnance Vol. XI, No. 62, p. 126*)

FIG. 8.—DEVELOPMENT OF WELDING TECHNIQUE AT WATERTOWN ARSENAL.

A. Photograph of a weld that judged by visual inspection was particularly good. B. X-ray picture through this weld. The black spots represent gas cavities. Judged by X-ray inspection the weld was particularly poor. C. X-ray picture taken after development of proper technique. This weld resembled the first one so closely that it would have been difficult to distinguish them by a visual examination. The metal of the weld was found to be sound. The weld was nearly perfect.

of welding practice, however, differs somewhat from the similar control of casting practice. In the latter case, such factors as proper size and location of gates and sinkheads, proper core strength, proper venting of sand molds, etc., are sought. These are fixed conditions, and once determined are not subject to variation. In the case of electric arc welding, the factors sought are proper size and type of welding rod, proper angle of holding the electrode, proper current and voltage in the arc, etc. Most of the factors are not fixed conditions, but are subject to variation by the operator during the operation.

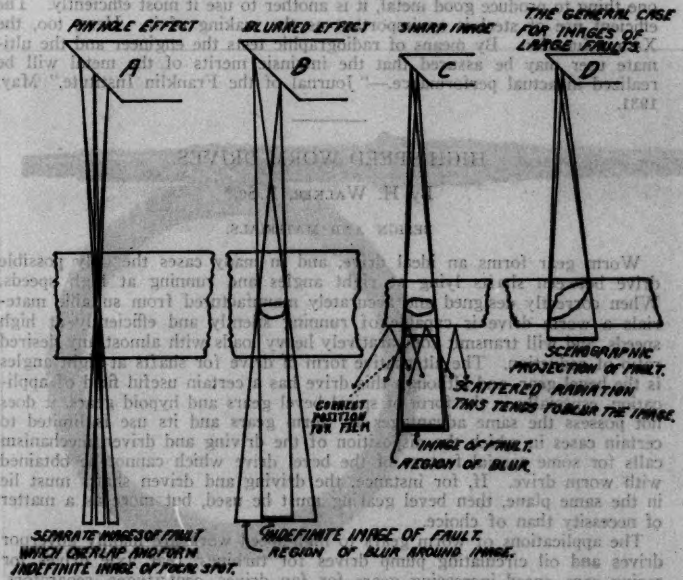


FIG. 4.—X-RAY PHOTOGRAPHY.

Schematic diagrams showing how defects in steel are photographed by X-rays. *A*, *B*, *C*, and *D* represent the anode of an X-ray tube. The cavity acts really as the aperture of a pin hole camera. When it is very small the image formed is really that of the active area of the anode surface. When it is larger an image of the cavity results.

For this reason X-ray control of welding technique is of more personal application. It forces the operator to be acutely conscious of his work at all times, and to give it the closest attention. It develops the welder. This is of great importance. It is not practicable, at present at least, to test the welds of a steel building other than by visual inspection. This inspection, however, is sufficient if it is backed up by a knowledge of the welding rod, the current and potential used, and a knowledge of the ability, and the character of the welder.

In the foregoing, the writer has attempted to point out in a general way that X-rays have become exceedingly important to the steel maker and user. Industry, which is vital to our civilization, depends on steel. It has been erected on a foundation in which steel is of dominant importance. The skeleton of its super structure is steel. Its future development depends on steel or a hypothetical steel substitute. Steel must be improved to meet the growing demands of a new age. This improvement, as far as it is possible to predict the future, will result largely from fundamental information that has been or will be obtained by means of X-ray diffraction studies. It is one thing to produce good metal, it is another to use it most efficiently. The efficient use of steel is as important as the making of it. Here, too, the X-rays function. By means of radiographic tests the engineer and the ultimate user may be assured that the intrinsic merits of the metal will be realized in actual performance.—"Journal of the Franklin Institute," May, 1931.

HIGH-SPEED WORM DRIVES.

By H. Walker, B.Sc.*

DESIGN AND MATERIALS.

Worm gear forms an ideal drive, and in many cases the only possible drive between shafts lying at right angles and running at high speeds. When correctly designed and accurately manufactured from suitable materials a worm drive is capable of running silently and efficiently at high speeds, and will transmit comparatively heavy loads with almost any desired ratio of reduction. The alternative form of drive for shafts at right angles is the bevel gear, and although this drive has a certain useful field of application, especially in the form of spiral bevel gears and hypoid gears, it does not possess the same advantages as worm gears and its use is limited to certain cases in which the disposition of the driving and driven mechanism calls for some special feature of the bevel drive which cannot be obtained with worm drive. If, for instance, the driving and driven shafts must lie in the same plane, then bevel gearing must be used, but more as a matter of necessity than of choice.

The applications of worm gears to high-speed work are mainly governor drives and oil circulating pump drives for turbines, rear axle drives for racing cars, speed increasing gears for fan drives, centrifuges, separators, pumps, &c., and certain special drives for machine tools. The worm speed in some of the above cases may be as high as 10,000 R.P.M., and even at this speed the gears will function satisfactorily, provided due precautions are taken in the design, manufacture and mounting of the gears. So far there has been found no practical application which limits the use of worm gear owing to high speed. Similarly, no application has been found which limits the use of worm gears owing to high peripheral speed. Rubbing speeds up to 6000 feet per minute are often met with in practice, and in exceptional cases much higher speeds are employed. The rubbing speed is the actual speed at which the worm thread surface passes over the wheel tooth, and is equal to the tangential speed of the worm thread resolved along the thread helix.

It will readily be appreciated that at such speeds as those quoted above there is no room for faults either in the materials or manufacture of the gears. A nine-thread worm running at 10,000 R.P.M.—see Figure 1—

* Technical Engineer, John Holroyd and Co., Milnrow.

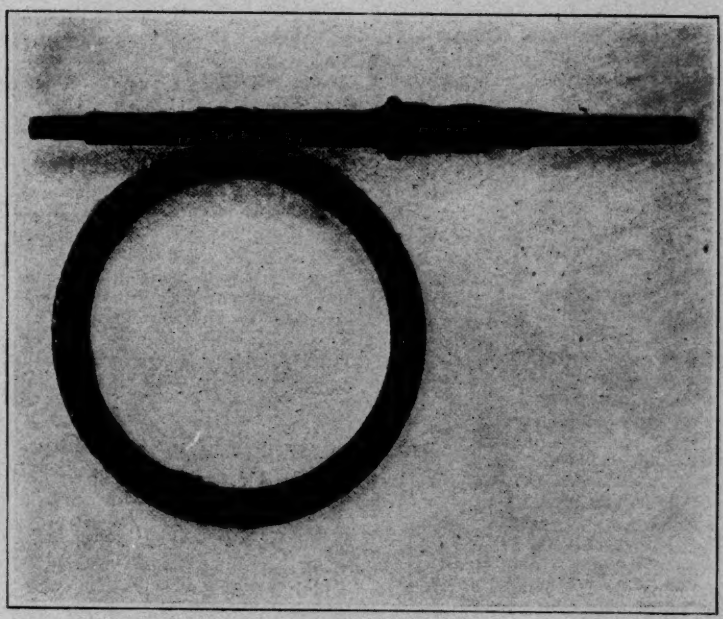


FIG. 1.—WORM DRIVE FOR 10,000 R.P.M.

means that 90,000 teeth are moving in and out of mesh every minute, or 1500 teeth every second. Inaccuracies in pitch will cause minute but rapid accelerations and decelerations of the worm wheel, resulting in vibration and noise. Such errors, combined with inaccuracies in mounting, or in tooth or thread profile, also result in faulty tooth contact; the tooth pressure, instead of being distributed over the theoretical lines of contact, is concentrated on certain parts of the tooth, increasing the specific loading and resulting in high local surface stresses which may in time cause pitting or rapid wear of the wheel teeth.

This question of accuracy is not one which depends solely on the skill of the workman and the accuracy of his machines. It depends also to a considerable extent on the design of the gears and the selection of a tooth form which lends itself to accurate generation. In designing the gears the object is primarily to make the center distance and general proportions such that the most compact drive is obtained whilst maintaining a suitable factor of safety. It will be appreciated that, although an increase in center distance and general proportions will reduce the specific surface stress tending to cause abrasion under a given load, this increase will also cause a corresponding increase in the rubbing speed of the gears. Since higher rubbing speeds call for lighter loads, it follows that in designing the gears a balance must be obtained such that the rubbing speed is kept as low as possible whilst maintaining a suitable factor of safety against wear of the teeth. This adjustment, of course, requires a considerable knowledge of the results obtained from gears in actual practice, but, especially when this knowledge is coupled with the results of laboratory tests on a testing machine such as the machine described later in this article, it is possible to determine accurately the required balance between rubbing speed and tooth pressure.

Having arrived at this stage, the question of thread form must receive consideration. In discussing this subject, reference will be made mainly to the thread form in its effects on comparatively low lead angle worms such as are met with in high-speed worm drives for governor and pump drives on turbines, these being the principal applications of high-speed worm drives. For such applications the worm has usually to be bored to fit on to a a large diameter shaft. The pitch diameter of the worm must therefore be large, and hence the lead angle must be low as compared with most other applications of worm drive.

The primary essential of a worm thread form is that when it is meshed with a suitable wheel the gears shall be theoretically capable of transmitting uniform angular velocity; that is to say, when the worm is turned at a uniform rate, it must be theoretically capable of propelling the wheel with uniform angular velocity and without intermittency of motion. There have been many conflicting opinions expressed as to whether worm gearing is really capable of theoretically fulfilling this condition. Generally speaking, the consensus of authoritative opinion is that worm gearing of the straight-sided linear type fulfils the condition, although one authority has stated that there is ondly one thread form which will fulfill the condition, the particular thread form referred to not being the straight-sided linear type. The truth of this matter is that there is an infinite number of thread forms which will give uniform angular motion.

In order to prove this statement it is only necessary to refer to the method of producing worm gears. The essence of the hobbing process for cutting worm wheels is that a hob, the successive positions of whose profile sweep out what corresponds to a complete envelope of the worm threads, is made to revolve with uniform angular velocity, whilst cutting the tooth spaces in

a wheel which also moves with uniform angular velocity. Assuming that the cutting of the teeth is carried out to perfection, then a perfect worm, gearing with this wheel, would reproduce the uniform relative velocities which obtained between the hob and the wheel, provided that the tooth proportions are such as to ensure overlap in the tooth engagement—that is, one tooth of the wheel must not be out of mesh before the next one comes into action. In order to show that this overlap of tooth engagement can be easily obtained it is only necessary to consider the tooth action on the central—axial—section of a straight-sided linear worm thread and wheel. In this case the worm section is similar to that of an involute spur rack, and the section of the wheel is similar to that of an involute spur pinion. Uniform rotation of the worm causes the thread profiles on the section under considertaion to advance at a uniform rate and the wheel, since it has conjugate teeth, must revolve uniformly. As in the case of an involute rack and pinion, overlap of tooth engagement on the central section is sufficient to impart uniform angular velocity to the gears. It is not in any way inherent to the argument that the worm profile shall be straight sided on the linear section. It may be a curve of any form provided it fulfils the condition that the wheel profile generated by this curve has successive normals which intersect the pitch line of the wheel at successive points. If this condition is not fulfilled interference will take place, and although this interference does not necessarily destroy the uniformity of angular velocity—the interference is cut away by the hob—it results in a loss of contact and consequently reduces the load-carrying capacity of the gears.

The same remarks hold good regarding the contact on sections other than the mid-section of a worm and wheel. Figure 2 shows two sections of a worm and wheel taken at right angles to the wheel axis. The central section is of the straight linear type, and the other section has been obtained in accordance with this proviso. In each case it will be seen that correct contact is obtained without interference, so that, although contact on the central section is sufficient to ensure uniform angular velocity transmission, additional contact is obtained on the offset sections, resulting in lines of contact which extend across the faces of the teeth. The actual formation of these lines of contact can be varied by using different thread shapes and by varying the thread proportions. Figure 4 shows the nature of the lines of contact on a typical low lead angle, straight-sided linear section worm. O O^1 is the axis of the worm, and Q Q^1 the axis of the wheel. The area within which contact takes place is bounded by the full lines passing through the points A, B, C, D, E, F, G. This surface is a distorted plane known as the "zone of contact," and the lines made by the intersection of the zone of contact with the thread surface are lines of contact. Any thread which lies within the zone of contact must have a line of contact where it intersects the plane. Such lines of contact are represented by a, b for one particular angular position of the worm. When the worm revolves the lines of contact move along the stationary zone in the direction of the arrow, and pass out of contact, being succeeded by other lines of contact.

The lower diagram in Figure 4 shows the actual tooth contact on a section of the worm wheel near the extremity of the wheel face, corresponding to the line E F on the contact zone. E^1 F^1 is the line of action or path of contact for the driving face under consideration, the apparent motion of the worm threads, and the real motion of the wheel teeth, being in the directions indicated by the arrows. It will be noted that on this section the pressure angle of the profiles on the driving side is comparatively low, and that the larger part of the contact takes place before the pitch point is reached. Both the low pressure angle on this section and the fact that the

contact takes place largely on the approach side of the pitch point contribute to make the straight-sided linear section worm deficient in load capacity.

It is well known that under heavy loads pitting of the wheel teeth starts on the leaving side of the worm wheel tooth. The fact that a low pressure angle occurs in this section means that the relative curvature of the profiles at the point of contact is high. Consequently, the "closeness of fit" of the contacting surfaces is low, thus increasing the specific tooth pressure for a given load, and increasing the possibility of failure through high surface stress. There is also a secondary contact which takes place on the leaving side of the wheel tooth, *i.e.*, certain points on the worm thread which have already been in contact on the entering side of the tooth, come into contact again on the leaving side, thus resulting in poor lubricating conditions on the leaving side.

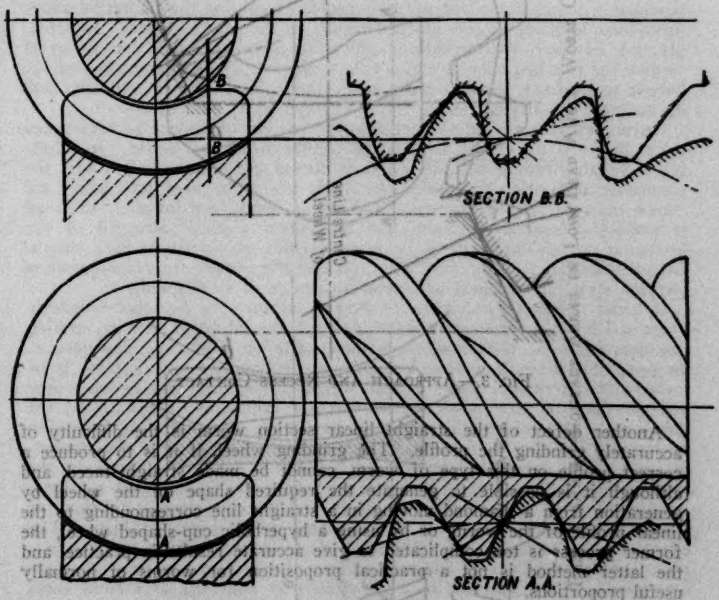


FIG. 3.—SECTIONS OF A WORM AND WHEEL.

The long approach to the pitch point results in increased sliding velocity between the teeth, in a direction at right angles to the tangential movement of the worm thread. This high sliding velocity on the leaving side of the tooth is a material addition to the tangential sliding component velocity, and in the past has been overlooked in estimating rubbing speeds. Moreover, the effect of a sliding velocity on the approaching side of the pitch point is conducive to greater frictional losses than if the contact were on the opposite side of the pitch point. Examination of Figure 3 verifies this fact. In Figure 3a, which illustrates "approach" contact, it will be seen

that the tooth force exerted on the wheel at the point contact is inclined away from the common normal and towards the center of the wheel. It is quite conceivable that if the pressure angle were high enough, the worm could not be made to drive the wheel, because the tooth force would act through, or beyond, the center of the wheel. Figure 3b, on the other hand, illustrates "recess" contact. The direction of sliding is now in the opposite direction, so that the tooth reaction is inclined as shown, and it is impossible to stop the gears from driving, no matter how high the pressure angle may be.

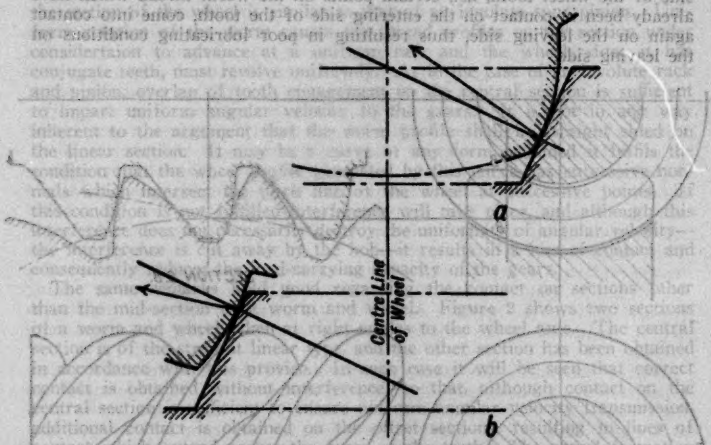
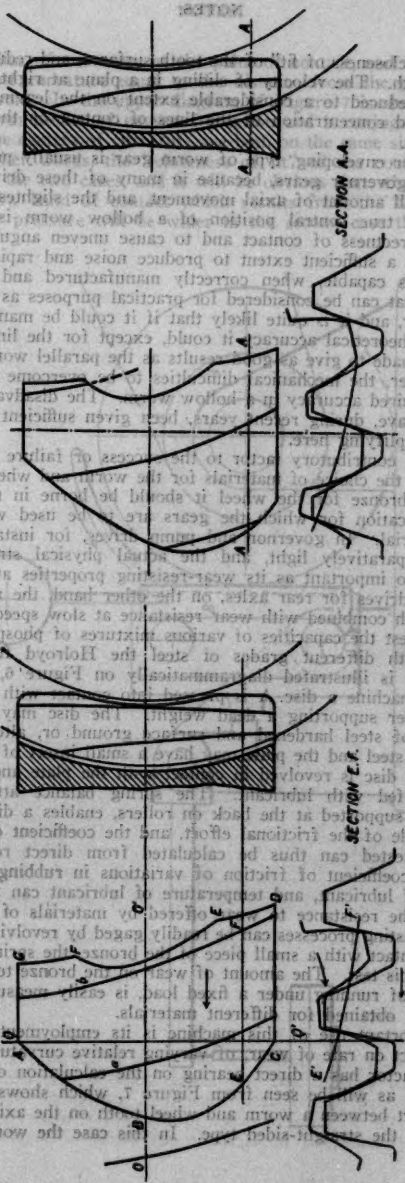


FIG. 3.—APPROACH AND RECESS CONTACT.

Another defect of the straight linear section worm is the difficulty of accurately grinding the profile. The grinding wheel, if it is to produce a correct profile on this type of worm, cannot be made straight-faced, and although it is possible to generate the required shape on the wheel by generation from a diamond moving in a straight line corresponding to the linear profile of the worm, or by using a hyperbolic cup-shaped wheel, the former process is too complicated to give accurate results in practice, and the latter method is not a practical proposition for worms of normally useful proportions.

In order to overcome these defects, the Holroyd process of worm thread design and generation was developed. The threads of the worm are ground from an abrasive wheel formed by a straight line movement, and by correct attention to the design and tooth proportions—mainly with regard to the position of the pitch plane of the worm in relation to the overall and root diameter of the threads—the difficulties of the straight linear worm are largely overcome, as will be seen from Figure 5, which shows the contact obtained on a pair of Holroyd gears of the same general proportions as those in Figure 4. It will be seen from Figure 5 that the contact at the leaving side of the tooth takes place largely on the recess side of the pitch point. The pressure angle on the leaving side is also increased, thus



FIGS. 4 AND 5.—CONTACT AREAS IN LOW LEAD ANGLE WORM GEARS.

increasing the "closeness of fit" of the tooth surfaces, and reducing potential wear of the teeth. The velocity of sliding in a plane at right angles to the wheel axis is reduced to a considerable extent on the leaving side of the wheel tooth, and concentration of the lines of contact on this side of the tooth is avoided.

The hollow, or enveloping, type of worm gear is usually precluded from use on turbine governor gears, because in many of these drives the worm shaft has a small amount of axial movement, and the slightest axial movement from the true central position of a hollow worm is sufficient to destroy the correctness of contact and to cause uneven angular movement of the gears to a sufficient extent to produce noise and rapid wear. The hollow worm is capable, when correctly manufactured and mounted, of transmitting what can be considered for practical purposes as true uniform angular velocity, and it is quite likely that if it could be manufactured and mounted with theoretical accuracy it could, except for the limitation above mentioned, be made to give as good results as the parallel worm. In actual practice, however, the mechanical difficulties to be overcome are too great to give the required accuracy in a hollow worm. The disadvantages of the hollow worm have, during recent years, been given sufficient publicity and do not need amplifying here.

An important contributory factor to the success or failure of high-speed worm drives is the choice of materials for the worm and wheel. In selecting a suitable bronze for the wheel it should be borne in mind that the particular application for which the gears are to be used will affect the choice of material. In governor and pump drives, for instance, the load is usually comparatively light, and the actual physical strength of the bronze is not so important as its wear-resisting properties at high speeds. In worm gear drives for rear axles, on the other hand, the requirement is physical strength combined with wear resistance at slow speeds.

In order to test the capacities of various mixtures of phosphor-bronze in combination with different grades of steel the Holroyd friction testing machine, which is illustrated diagrammatically on Figure 6, was devised. In using this machine a disc A is pressed into contact with a plate B by means of a lever supporting a dead weight. The disc may be of bronze and the plate of steel hardened and surface ground or, alternatively, the disc may be of steel and the plate may have a small insert of the bronze to be tested. The disc is revolved in contact with the plate and the contacting parts are fed with lubricant. The spring balance attached to the plate, which is supported at the back on rollers, enables a direct measurement to be made of the frictional effort, and the coefficient of friction for the materials tested can thus be calculated from direct readings. The effects on the coefficient of friction of variations in rubbing speed, materials, quality of lubricant, and temperature of lubricant can thus be easily determined. The resistance to wear offered by materials of various compositions and casting processes can be readily gaged by revolving a hardened steel disc in contact with a small piece of the bronze, the spring balance not being used in this test. The amount of wear on the bronze test piece, after a certain time of running under a fixed load, is easily measured and comparative results obtained for different materials.

Another important use of this machine is its employment to determine the precise effect on rate of wear, of varying relative curvatures of the test pieces. This factor has a direct bearing on the calculation of the stresses in worm gears, as will be seen from Figure 7, which shows in the lower view the contact between a worm and wheel tooth on the axial section, the worm being of the straight-sided type. In this case the worm thread has

an infinite radius of curvature at the point of contact, and the wheel tooth has a radius of curvature equal to the distance from the point of contact to the base circle. In the upper diagram of Figure 7 is shown the contact between a worm profile of curved formation and a conjugate wheel tooth. In this case the centers of curvature are both on the same side of the worm profile. In other words, the "closeness of fit" of the contacting surfaces is different in the two cases, and it will be obvious that when placed under load the narrow band of contact which results from the flattening of the surfaces under pressure will be wider in the second case than in the first

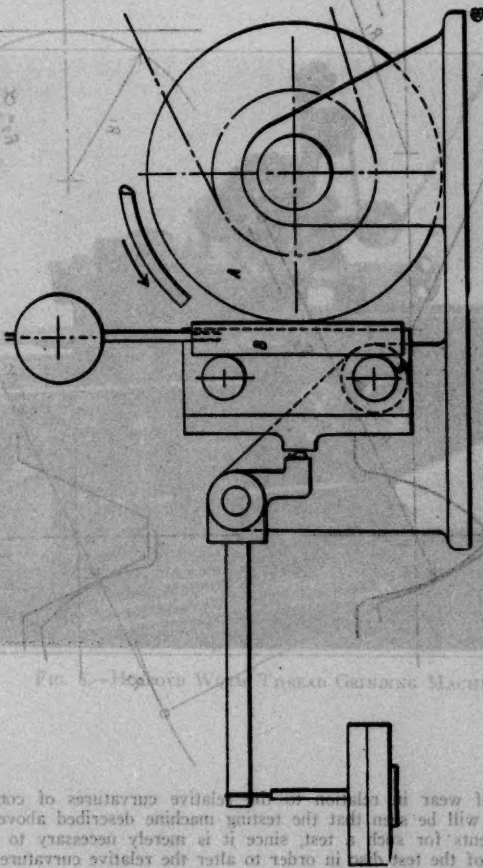


FIG. 6.—HOLROYD FRICTION TESTING MACHINE.

case. It is well known that surfaces wear better when they fit more closely. The improved wearing properties of a 20 degree pressure angle spur gear as compared with a 14½ degree spur gear are directly attributed to the increased closeness of fit obtained by increasing the pressure angle. So far as the writer is aware, however, no direct measurements have ever

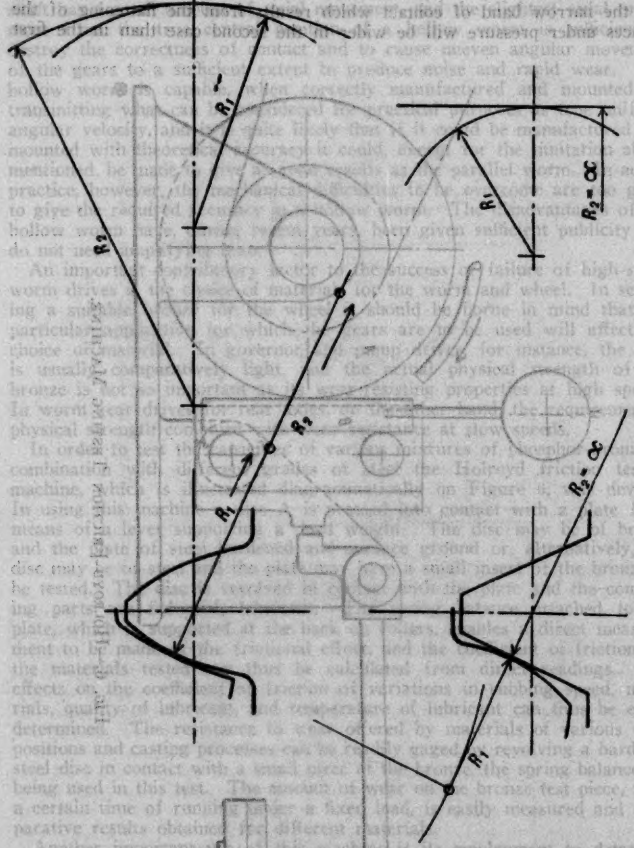


FIG. 7.—CLOSENESS OF FIT IN WORM GEARS.

been made of wear in relation to the relative curvatures of contacting surfaces. It will be seen that the testing machine described above fulfils the requirements for such a test, since it is merely necessary to change the diameter of the test disc in order to alter the relative curvature at the point of contact, and thus to ascertain the effect of such an alteration.

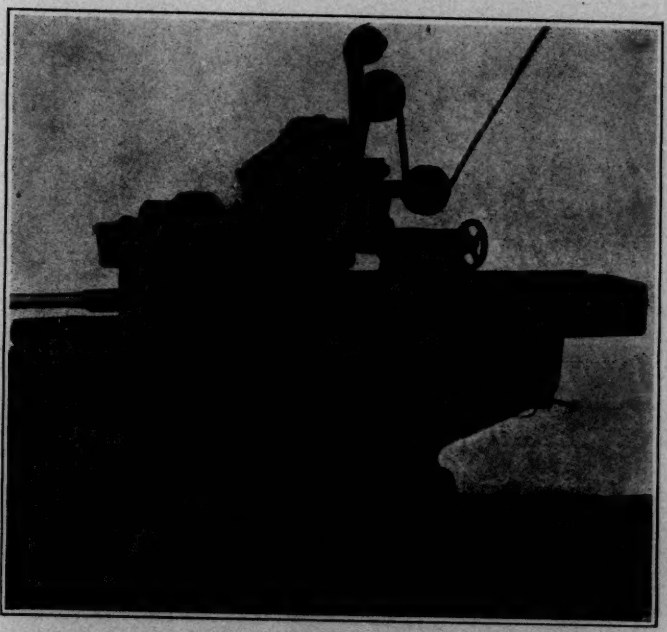


FIG. 8.—HOLROYD WORM THREAD GRINDING MACHINE.

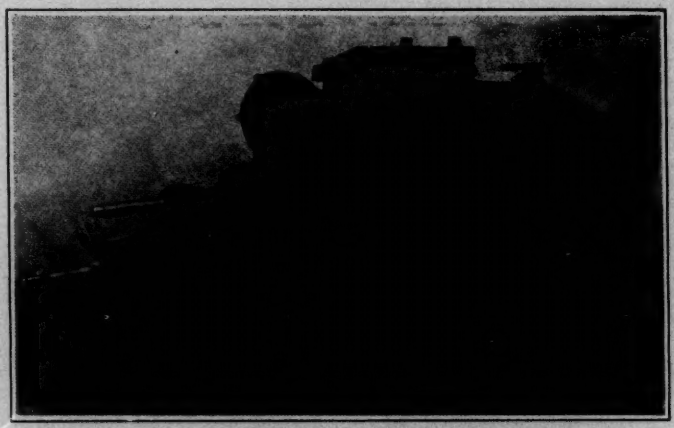


FIG. 9.—HOLROYD WORM WHEEL GENERATING MACHINE.

FIG. 11.—WORM AND WHEEL TESTING DEVICE.

For the purposes of this particular test the flat surface against which the disc presses must be kept moving, for otherwise a small hollow would be worn in the plate and the relative curvature would thus change during the test from the original value.

THE MANUFACTURE OF HIGH-SPEED WORM DRIVES.

Ranking equally important with questions of design and materials is the manufacture of the gears. If the full theoretical advantages of a gear are to be obtained in practice it is essential that accuracy of manufacture should be maintained. The effects of inaccuracies in the gears are: (1) The theoretical contact between the teeth is not obtained and the result is that certain "high spots" on the teeth have to carry the entire load. This effect results in heavy local surface stresses which tend to cause rapid wear or pitting of the teeth. The nearer the approach to theoretical accuracy the better will be the distribution of the load, and, of course, a low specific load results in less wear of the teeth. (2) Inaccuracies cause vibration and noise. Both these effects are particularly objectionable when emanating from worm gears fitted to so silent a unit as a modern turbine, besides being a general indication of inefficiency.

The following notes regarding the manufacture of the gears are therefore made with reference only to those operations which control the accuracy, as it is not considered necessary to refer to the normal operations of turning, thread milling, &c.

Between the worm and the wheel, it is on the worm that the highest degree of accuracy is required, because any wear which takes place on the wheel will tend to eliminate inaccuracies, whereas if the inaccuracies exist in the worm they cannot be eliminated by running the gears. The operations which govern the final accuracy of the gears are the worm thread grinding and the generating of the wheel teeth. The worm thread grinding operation is, for the reason given above, the more important. It is, moreover, more difficult to ensure accuracy in the worm than in the wheel, because of the numerous sources of error which can occur in a thread grinding machine. It was with a realization of these facts that the Holroyd thread grinder, illustrated in Figure 8, was designed. This machine was built with the express intention of using it for grinding the threads of worms for turbine governor drives, and in order to maintain its accuracy is used for no other purpose.

So far as the general principle of the machine is concerned, it follows the usual design of a thread grinder in that the slide carrying the revolving worm is traversed past the grinding head by means of a lead screw, but it differs in several other respects. The amount of gearing used in the machine is reduced to a minimum. It has no differential, because this item increases inaccuracies. The few gears which are used in the main drives of the machine take the form of spur gears, hardened and ground on the tooth profiles. For certain leads, hardened and ground spur wheels are available for change gears, and the lead screw itself is specially corrected for pitch error.

The dividing at the end of each stroke of the grinder is carried out by means of a dividing plate with a single slot. This dividing plate is driven by means of a small auxiliary motor, and motion is transmitted from the dividing plate, through change gears, to the lead screw nut which is attached to the slide. There is a main motor for driving the work and the lead screw. This motor automatically reverses after each grinding stroke, to bring the slide back to its initial position, and is automatically cut out of

operation when the dividing motor is working. The main motor drives through a hydraulic variable speed mechanism, which enables the forward or return movement of the table to be varied immediately by turning one of two handles. Since the dividing and lead screw motions are separate and self-contained it is easy to detect and remedy any errors which may creep into the machine, and the results can, therefore, be kept constantly up to a standard.

The cutting of the wheel teeth is carried out on a Holroyd worm wheel generator similar to that shown in Figure 9. This machine employs a relatively large worm wheel for driving the table. Any errors in the dividing worm wheel driving the table are reproduced on the worm wheel blank, but are reduced in proportion to the relative sizes of the worm wheel blank and the table-driving worm wheel. It is, therefore, important on a machine of this type to keep the table-driving worm wheel as large and accurate as possible. The machine has a tangential feed which causes the hob slide to travel tangentially past the worm wheel blank, and a differential is incorporated to compensate the feed motion.

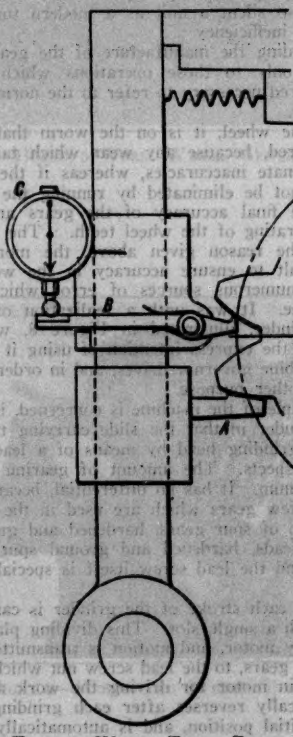


FIG. 10.—WHEEL TOOTH PITCH MEASUREMENT.

The remaining factor which controls accuracy of production is the accuracy of the cutting tool for the worm wheel. In cases involving the production of quantities of similar gears a full hob is made, and in order to eliminate hardening errors the hob is profile ground, using a special grinding attachment on a relieving lathe. In cases in which the diversity of size does not justify the manufacture of a hob, a single-point, or fly, cutter is used. It is important that such a cutter should be ground in position in the cutter bar, and that it should preferably be ground on the worm grinding machine, using the same setting as is used for grinding the worm, in order to ensure exact similarity between the worm profile and the cutter profile, relief being given to the cutter afterwards.

TESTING.

Apart from routine inspection, the important factors to be tested in a pair of worm gears are the accuracy of spacing of the wheel teeth, the spacing of the worm threads, on both a linear and a transverse section, and

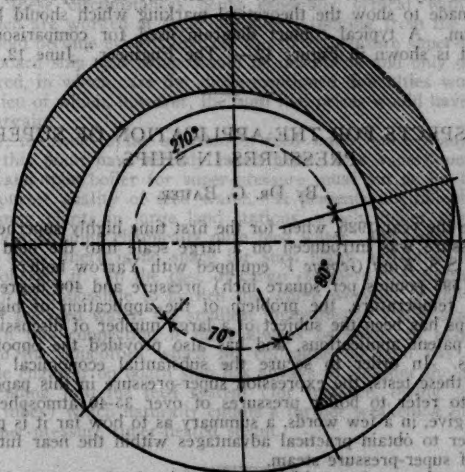


FIG. 12.—THEORETICAL CONTACT DIAGRAM.

the profiles of both worm and wheel. The pitch errors in the wheel teeth are detected by comparative measurements from tooth to tooth, using an instrument diagrammatically illustrated in Figure 10. A tooth of the wheel, which is mounted on an arbor, is brought into contact with the fixed stop A. The dial indicator C is then set at zero with the lever B in contact with the next tooth of the wheel. Successive readings are then taken between each tooth and its adjacent tooth, and variations in pitch are indicated on the dial. It will be noted that these readings give comparative errors only, but they can be reduced to give absolute errors by taking the algebraic sum of the dial readings once round the wheel, divid-

ing this by the number of teeth in the wheel, and adding the result algebraically to the individual dial readings. A similar method is used for testing errors in the worm, but in this case the dial indicator is moved axially, from thread to thread, through a distance equal to the linear pitch. This distance is measured by slip gages and a positive reading is, therefore, obtained immediately.

Profile errors can be measured by a gage applied to the worm threads, but because of the difficulty of measuring the wheel profile, the best method is to gage the combined accuracy of both the worm and wheel profiles by observation of the "marking" obtained on the teeth and threads when meshed at the correct centers. This test is carried out by mounting the gears at correct centers on the fixture shown in Figure 11. The worm threads are marked lightly with Prussian blue, and the effect of running the gears round by hand is to show very clearly on both worm and wheel where contact is taking place. Much more information is obtained from the marking on the worm than that on the wheel, and usually an examination of the worm contact is sufficient to show the position and approximate magnitude of any errors. In order to facilitate the inspection, a diagram is usually made to show the theoretical marking which should be obtained on the worm. A typical contact diagram used for comparison with the actual result is shown in Figure 12.—"The Engineer," June 12, 1931.

THE PROSPECTS FOR THE APPLICATION OF SUPER STEAM PRESSURES IN SHIPS.

By Dr. G. Bauer.

Ever since the year 1926, when for the first time highly superheated steam of high pressure was introduced on a large scale into the field of marine drive in the S.S. *King George V* equipped with Yarrow boilers of 40.5 atmospheres (595 pounds per square inch) pressure and 400 degrees C. (752 degrees F.) temperature, the problem of the application of high-pressure steam in ships has been the subject of a large number of discussions, calculations and patent applications, and has also provided the opportunity for various tests. In order to secure the substantial economical advantages revealed by these tests, the expression super-pressure in this paper shall be understood to refer to boiler pressures of over 35-40 atmospheres. It is proposed to give, in a few words, a summary as to how far it is possible for the shipowner to obtain practical advantages within the near future by introduction of super-pressure steam.

Anticipating the result at which I shall arrive in the course of this paper, I should express my opinion to the extent that for the present, and in all probability in the near future, it will be well, as far as the marine drive is concerned, to confine oneself to trials with partial installations which are to be carried out carefully, and in the meantime to aim at the improvement of economy by taking other ways which entail lower risks but are almost equally successful. Here the objection will be raised that it should easily be possible to adapt technical innovations for use on ships, which, as is the case with the super-pressure steam, have already made considerable progress in land plants, and now and again at this issue a word concerning the traditional slowness of marine engineers will be heard.

In reply to such objections I should point out that especially in the case of boiler work, the circumstances on board ship are entirely different from those at stationary plants for the following reasons:—

Firstly, in land plants there are no special circumstances providing difficulties to use water-tube boilers which alone are suited for the generation of super-pressure, because it is possible, if coal is used as a fuel, to work either with mechanical stokers or with calibrated coal of a certain quality or with pulverized-coal fuel. It may be assumed to be generally known that on ships the mechanical stokers as well as the pulverized-coal fuel are, in spite of a number of partial successes which have been obtained, far from being capable of common or even extensive adaptation, and have progressed only very slowly during recent times, although many trials and efforts have been expended. The impossibility of finding in every port of the world suitable brands of coal, the difficulty of accommodating the bulky and not easily governed apparatus on board and the great power consumption, are things which still provide a great number of problems for solution. On the other hand, it would be impracticable to use water-tube boilers without mechanical stokers or equipment for pulverized fuel which in the old-fashioned manner are simply stoked by hand with coal, because of the difficulties of cleaning the large grates and the tubes, also because of the uneven combustion, and finally because of the danger of burning the boiler doors and floor plates on board, all these circumstances being especially critical with water-tube boilers which are worked at super-pressure. Judging from the above viewpoint it would become evident that for the super-pressure super-heated steam boiler on board ship coal firing would be unsuitable and only oil firing can be considered, in which case the above-mentioned difficulties would be eliminated, in lieu of which, however, the high fuel prices would have to be taken into the bargain.

Secondly, to come to the question of feed water, since many discussions regarding this point have already taken place, it is superfluous to remark that the water-tube boiler for super-pressure must put as its first requirement the purest quality of feed water. It presents no difficulty to install water-cleaning plants in large land stations, which, under careful control, supply almost absolute chemically pure feed water. On board ship, even in high-class ships, there is no space for such plant, and often, not the personnel to operate it. In the case of cargo steamers, the circumstances are such that it is impossible to burden the whole working of the engine with the care of the feed water, the smallest impurities in which are highly dangerous. In this connection it must always be borne in mind that quite apart from the difficulties which his colleague on land has in this connection, the engineer on board ship has always to consider the danger of sea water intruding into the feed system (for instance, owing to leakage of condenser tubes) which naturally entails a further weakening of the position against super-pressure super-heated steam.

MANEUVERING.

Thirdly, the question of maneuvering must be dealt with. The land boiler is equipped for an output which continuously remains uniform, or for a rise of output which is known in respect of its time and extent, and the boiler can leisurely be switched on or off, whereas as quantity of steam drawn from a marine boiler varies widely, especially when maneuvering. Frequently its supply is suddenly entirely interrupted, and equally suddenly it is again pushed up to the highest maximum attainable. It follows that super-pressure boilers consisting only of tubular systems and drums of very small diameter (which is necessary in order to obtain the highest possible stability against internal pressures) are for working on board only very little suited unless special precautions are taken, whereas in land plants there are no objections against the use of systems of the above kind. The use on ships of super-pressure boilers with their small water space may, in

special circumstances when maneuvering, result in the boilers being so far deprived of their contents that the water tubes exposed to the fire will at least be unduly heated and damaged even if their burning through can at all be avoided. One is therefore in the uncomfortable position, when applying super-heated presure steam to marine drives, that one cannot renounce the arrangement of water spaces of sufficient contents; on the other hand, however, one is unable to provide such for reasons of stability. In this connection it should be pointed out that in consequence of the small water level surface in the drums it cannot be avoided that impurities are torn away with the turbulently rising steam bubbles into the main pipe line, which again points to the great importance which must be attached to the most careful composition of the feed water when using super-pressure boilers with their necessarily small drums.

In addition it may be mentioned that, besides the working difficulties as described above, there are several other questions for which a satisfactory solution has not yet been found, which are obstructing the introduction of super-pressures on board ship. One of these questions is that of suitable material. It appears as though steam of high tension which has for itself a very high saturation temperature, and in super-pressure boiler plants is naturally subjected to the utmost superheating, has certain influences on the structure of the steel and tube material which are likely to cause breakdowns in special circumstances. There are, firstly, deposits inside the tubes which lead to the temperature of the tube walls approaching a certain limit at which the strength of the material offers insufficient resistance to the stresses acting thereon, or, secondly, erosions which unduly weaken the tube walls. To talk of constructive questions, there are several items which are still almost unsolved problems, as, for instance, the *per se* difficult manufacture of the drums where rivets must be avoided, the question of the covers, pipe connections, flange, tightenings and fastening of the boiler fittings, water-level gages and so forth.

Although admittedly very skilful solutions for the construction of the auxiliaries required for the use of super-pressure steam have been found, it must always be considered that a super-pressure plant requires several complicated auxiliaries in addition to those needed in a plant working under normal pressure. Firstly, it may be pointed out that the feed pumps are considerably larger and more intricate than at normal pressure. As is known, the capacity of feed-water pumps grows in proportion with the boiler pressure; on the other hand, it decreases relative to the feed-water consumption. For instance, in the case of a boiler plant which, like the Benson system, works at a pressure of 230 atmospheres, the power required for the drive of the feed pumps is raised to 15 times that which is required at 18 atmospheres, whereas, owing to the steam consumption being reduced by, say, 10 per cent, the capacity is only reduced by 10 per cent, *i.e.*, the output still remains 13.5 times as large as with a boiler plant under normal pressure.

It is understood that with the feed pumps also the feed pipes, feed valves and purifiers grow considerably more complicated, and their construction becomes more difficult. In addition, there is the apparatus for the most careful purification of the feed water, as mentioned above, the more difficult construction and tightening of the feed-water preheater, and the design of the fittings in which the question of water-level gages, which, at any rate, still presents a difficulty, has to be considered.

The greatest restriction, however, is imposed on the application of super-steam pressure for cargo vessels or less valuable passenger ships by the question of the choice of a suitable engine-room staff. Although the larger shipping companies have an extremely well-trained staff, it will be necessary even for them to use the best and most experienced hands on their high-class passenger vessels, so that for the cargo vessels only the younger and

A.	B.	C.	D.	E.	F = D − E.	G.	H.	I.	K.	L = I + K.	M = L − G
No.	Steam.		Total Heat.	Feed-Water Temperature.	Heat to be Generated in the Boilers.	Additional Generation as Against 1.	Adiabatic Heat-drop.	Increase as Against 1.	Addition on Account of Superheat	Available gain in heat-drop as against 1.	Total gain as against 1.
	Pressure.	Tempera-ture.									
—	Atm. Abs.	C.	Calor. per kg. Steam.	C.	Calor. per kg. Steam.	Per cent.	Calor. per kg. Steam.	Per cent.	Per cent.	Per cent.	Per cent.
1	16	300	726·0		576·0	0·0	223·5	0·0	0·0	0·0	0·0
2	16	350	752·0	150	602·0	4·6	236·2	5·7	+2	7·7	3·1
3	16	400	777·3		627·3	8·9	249·0	11·3	+4	15·3	6·4
4	19	300	723·0		573·0	−0·5	228·3	2·1	−0·2	1·9	2·4
5	19	350	750·5	150	600·5	4·2	241·5	8·0	1·8	9·8	5·6
6	19	400	776·0		626·0	8·6	254·2	13·8	3·8	17·6	9·0
7	30	300	716·5		566·5	−1·6	239·0	6·9	−0·7	6·2	7·8
8	30	350	745·5	150	595·5	3·4	253·2	13·2	1·3	14·5	11·1
9	30	400	772·0		622·0	8·0	267·0	19·3	3·3	22·6	14·6
10	60	300	687·5		537·5	−6·6	246·2	10·2	−3·7	6·5	13·1
11	60	350	728·5	150	578·5	0·4	266·3	19·6	−1·8	17·8	17·4
12	60	400	760·5		610·5	6·0	283·3	26·6	−0·2	26·4	20·4
13	120	—	—	—	—	—	—	—	—	—	—
14	120	350	675·5	150	525·5	−8·8	259·0	15·8	−6·8	9·0	17·8
15	120	400	730·5		580·5	0·8	287·5	28·5	−4·7	23·8	23·0
16	224	—	—	—	—	—	—	—	—	—	—
17	224	—	—	—	—	—	—	—	—	—	—
18	224	400	637·0	150	487·0	−15·4	252·5	13·0	−10·0	3·0	18·4

TABLE SHOWING TOTAL HEAT, HEAT-DROP, AND SAVING IN STEAM CONSUMPTION FOR VARIOUS PRESSURES AND TEMPERATURES.

less experienced staff is available, of whom one cannot yet, and still for several years hence, require to run the great risk involved in driving a super-pressure plant.

In consideration of these circumstances it is no wonder that there is only a very limited number of trial plants for super-pressure in marine drives as against its comparatively rapid and extensive introduction into land plants. This becomes the more intelligible if one takes a brief glance at the limited economical advantages of super-pressure as applied to ships' plants (I do not here refer to the question of high superheating), as will be done in the following. Looked at from a theoretical standpoint, the economical gains which can be obtained by raising the boiler pressure will be seen from the table on page 489. This table shows for various steam pressures and temperatures the heat necessary for their generation, the adiabatic drop (calculated down to a vacuum of 95 per cent,) and the saving in steam consumption which can be obtained as compared with a pressure of 15 atmospheres. We see from the table that at a superheated steam temperature of, *e.g.*, 350 degrees C. (662 degrees F.), the theoretical fuel saving gained by raising the steam pressure from 18 atmospheres to 60 atmospheres amounts to only 11.8 per cent, whereas at a constant tension of 18 atmospheres the raising of the superheating from 300 degrees to 400 degrees C. (572 degrees to 752 degrees F.) secures a practical saving of 6.6 per cent.

It will be seen from this that the successes obtained by the introduction of super-pressure superheated steam can to a considerably larger extent be ascribed to the superheated steam than to the super-pressure steam. At the same time, the introduction of the first mentioned, *i.e.*, the raising of the temperature of superheating in marine drive, offers, as we have seen, less difficulties than the raising of the boiler pressure.

Better than is possible by explanations of a general character, the question of the fuel saving, attainable by the introduction of super-pressure steam into ships' plants, can be made clear by an example. For this purpose let us choose the engine plant for a cargo vessel of 10,000-12,000 tons having an output of 5000 I.H.P. We will investigate this engine plant with a view to its fuel consumption, looking firstly into a case where the boiler plant consists of so-called Capus boilers (the well-known combination of cylindrical and water-tube boilers, as used on the S.S. *Ile-de-France*), the engine plant being a combined reciprocating engine with Bauer-Wach exhaust steam turbine arranged behind; secondly, into a case where Benson boilers and an engine plant consisting of geared turbines are used. The steam pressure before the main valve of the reciprocating engine or the high-pressure turbine respectively may be assumed to be 18 atmospheres or 60 atmospheres* respectively, and both cases shall be based upon a steam temperature of 350 degrees C. (662 degrees F.) and an exhaust pressure in the condenser of 0.05 atmosphere absolute.

MECHANICAL DETAILS.

Looking firstly into the case of the super-pressure plant, the turbine set of which may be designed in such a way that one pinion of the gear is driven by a high-pressure Curtis wheel running at 5000 R.P.M. with a drum behind it, the other one by a low-pressure turbine, the gear ratio obtained by a double gearing resulting in a reduction to 80 R.P.M. at the propeller shaft. For the calculation of this plant a mean blading diameter of the two staged high-pressure Curtis wheel of 660 millimeters has been assumed, the pressure ranging from 61 atmospheres down to 20 atmospheres, corresponding to a

* The calculation for the case that the steam enters the main admission pipe to the turbine with a pressure of 120 atmospheres has also been investigated; it however results in a very small advantage, approximately 2.5 per cent, against the above assumption.

heat difference of 61.5 calories. At a total steam consumption of 17,800 kilograms, a figure which can only be an approximate basis, an output of the Curtis wheel of 1160 H.P. at an efficiency of 67 per cent is obtained. It is to be noted here that the radial opening of the inlet nozzles is only 8 millimeters, in spite of which the percentage of steam admission into the area amounts to only 12.3 per cent. Naturally, these proportions work out rather unfavorably with a view to the ventilation loss, which reaches the figure of 100 H.P.

The steam is utilized in the high-pressure drum down to a pressure of 3 atmospheres, so that here a heat drop of 84.5 calories is available, which, taking into account the considerable clearance and eddy losses and the losses in the stuffing boxes (7 + 4 = 11 per cent), only means a utilization at a degree of efficiency of 72.5 per cent. With this degree of efficiency an output of the drum of 1720 H.P. is obtained. At the stages amounting to 27, the mean blading length of the drum would be only about 40 millimeters at a drum diameter of 300 millimeters and a length of the first blade of only 15 millimeters. These dimensions give a clear picture of the rather unfavorable circumstances given for the design of such turbines and the difficulties in the way of their construction, as far as economical performance is concerned.

From the high-pressure drum the steam, thereby incurring a small loss, passes into the low-pressure turbine, where it enters a conical drum at an initial tension of 2.8 atmospheres, the diameter of the drum beginning at 500 millimeters and ending at 685 millimeters. The moisture of the steam amounts to 4.5 per cent at the inlet and to 13 per cent at the outlet of the drum. The length of the first blade is, in spite of the small drum diameter at this place, not more than 35 millimeters, the last blade attaining a length of 170 millimeters. In consideration of the clearance losses of about 5 per cent, as well as the stuffing-box and eddy losses, and, furthermore, the necessary deduction for the moisture of the steam of 5 per cent and at last the outlet losses of 3 per cent, a degree of efficiency of the low-pressure drum of not more than 59.5 per cent can be counted upon, so that its output may be determined from the adiabatic heat drop of 132.5 calories to be 2590 H.P. The addition of the output of the high-pressure Curtis wheel and the high and low pressure drums results in a total output of 5470 H.P. From this the friction and ventilation work must be deducted, totaling about 250 H.P. If we assume the degree of efficiency for the gearing to be 96 per cent at this comparatively small output, then an effective total output of 5000 S.H.P. and a steam consumption of 3.56 kilograms per S.H.P.-hour is ascertained. This figure is in accordance with our initial assumption.

Passing over to the calculation of fuel consumption for the plant in question, we may base our investigations on coal having a heating value of 7500 calories. The preheating of the feed water is effected firstly by the exhaust steam of the engine auxiliaries amounting to 19 per cent in this case, because of the high output of the feed pumps for the steam used in the main engines, and secondly, by the exhaust steam of the ship's auxiliaries. Thus the feed water temperature is raised to approximately 150 degrees C. (302 degrees F.). Assuming that for the Benson boiler, as used in this case, coal firing as per the above example is possible, then we will not be much wrong in counting on a degree of efficiency of 80 per cent. The heat value of the high-pressure steam ready to work is 732 calories per kilo of which 150 are provided by preheating, so that there remain 582 calories to be imparted on the water or the steam in the boiler. This involves, basing on the above assumption, a 10.32 fold vaporization, and in consideration of the addition of 19 per cent for auxiliary purposes of which, as we saw, the feed pump requires a considerable part, namely, 7 per cent, the consumption of coal*

* For fuel oil having a lower calorific value of 10,000 calories the figure is 0.29 kilograms per B.H.P.-hour.

amounts to $\frac{3.56 \times 1.19}{10.32} = 0.411$ kilograms per B.H.P.-hour.

Turning now to the combined plant we choose a three-cylinder reciprocating engine having cylinder diameters of 780 millimeters (30.7 inches) for high pressure, 1250 millimeters (49.2 inches) for intermediate pressure, and 2000 millimeters (78.8 inches) for low pressure at a stroke of 1400 millimeters (55.2 inches) the propeller revolutions being 80 R.P.M. as in the case discussed. The steam leaves the reciprocating engine at a pressure of 0.6 atmospheres, and enters the exhaust steam turbine at a low tension loss, the exhaust steam turbine working at the back pressure of 0.05 atmospheres as fixed above for both plants. The number of revolutions of the exhaust turbine, fitted with 9 reaction stages, is 2650 R.P.M. Since a large number of similar plants have been constructed, the working results of which, based on careful readings, are plentifully available, the calculation of the steam consumption can be dealt with in a simple way.

The adiabatic fall available amounts to 240 calories, the total degree of efficiency of the reciprocating engine in I.H.P. being 80 per cent of that of the turbine, in S.H.P. 62 per cent. This figure converted into I.H.P., based on a mechanical degree of efficiency for the reciprocating engines of 91 per cent, rises to 68.2 per cent since the part of the adiabatic heat drop for the reciprocating engine is 161.3 calories, and the part to be taken by the turbine amounts to 84 calories (taking into account a low reheating), a total utilized heat drop hi of 186.5 calories based on the degrees of efficiency as assumed above results, entailing a steam consumption per I.H.P. an hour of

$$Di = \frac{632}{hi} = 3.39 \text{ kilograms per I.H.P. per hour.}$$

This steam amount expressed in effective horsepower results in $De = 3.39 : 0.91 = 3.72$ kilograms per B.H.P.-hour.

In order to ascertain the fuel consumption of the plant from the steam consumption of the main engine, we must first investigate how many calories must be applied to the preheated feed water in the boiler in order to attain the heat content of the steam of 756 calories corresponding to the primary stage of 18 atmospheres and 360 degrees C. (680 degrees F.*). If we assume the condensate temperature to be 25 degrees C. (77 degrees F.), the utilization of exhaust steam from the engine auxiliaries—amounting to approximately 13 per cent of the amount required for the main engine—is only sufficient to attain a preheating to approximately 110 degrees C. (230 degrees F.). If one added the exhaust steam of the ship's auxiliaries, one would obtain approximately 125 degrees C. (257 degrees F.). Therefore, in order to arrive at a feed water temperature equal to that of the super-pressure plant, we take from the reciprocating engine behind the high-pressure cylinder 3.6 per cent of the main engine steam which, since the high-pressure cylinder has already utilized approximately ¼ of the main energy, involves a rise in steam consumption of the main engine of $\frac{3}{4} \times 3.6 = 2.7$, *i.e.*, from 3.72 — 3.82 per cent.

Assuming a degree of efficiency of 80 per cent for the Capus boiler at ample air preheating based on the above statements, the vaporization is

$$\frac{7500 \times 0.8}{756 - 150} = 9.9 \text{ fold, } i.e., \text{ at a steam consumption of 3.82 kilograms per}$$

* If this calculation were carried out for the usual cylindrical boilers working at 15 atmospheres, the fuel consumption figures arrived at with a superheating temperature of 330 degrees C. (626 degrees F.) would be more unfavorable by 4 per cent compared with the present case.

S.H.P. hour, the coal consumption for the S.H.P. hour is $\frac{3.82 \times 1.13}{9.9} = 0.436$ kilograms,* or for the I.H.P.-hour 0.397 kilograms.* It follows that the combined plant working at normal pressure is surpassed in coal consumption by only 5.8 per cent by the high-pressure plant.

A viewpoint which is not to be neglected when making economical comparisons of the above kind is that also at a relatively considerable percentage of saved fuel, the absolute value of saved fuel decreases with the proportion of the rise in economy already obtained. For instance, looking at the above example, the saving in fuel consumption obtained by use of super-pressure would be 5.7 per cent of 0.4 kilograms, *i.e.*, 0.0228 kilograms per I.H.P.-hour, or for an output of 5000 H.P., 0.114 ton per hour, resulting in a saving of 570 tons of coal if the ship works for 5000 hours per year, which would, based on an average coal price of £1, be a total saving of £570. It will easily be seen that neither the purchase price nor the upkeep of the super-pressure plant can be allowed to be very considerable to make the installation of such a plant, with all its risks and dangers, economically worth while.

In connection with super-pressure plants the question of intermediate superheating is much discussed. At the use of very high pressures the superheating in the super-heaters combined with the boiler plant cannot be raised so far (an obstacle to this is the tensile strength of iron at high superheating temperatures) that the adiabatic expansion in the low-pressure end of the main engine cannot be prevented from intruding far into the range of wet steam, whereby the efficiency of the main engine suffers and also otherwise various difficulties are caused. The idea of providing intermediate superheating is therefore put forward. Here, however, one encounters the difficulty that from a designing standpoint it is almost impossible to lead the steam from the engine room again to the boiler room, and, after intermediate superheating, back into the engine room, so that it is only left to use live steam for intermediate superheating. The installation of apparatus of this kind, however, entails complications and expense in the engine plant, against which the advantages thereby obtained are of no practical importance. An exact calculation for the turbine plant of our example shows practically no gain at 60 atmospheres initial pressure, nor at 120 atmospheres initial pressure.

As regards the feed water preheaters, it seems tempting with super-pressure plants to raise considerably the feed water preheating as against that of plants working at normal pressure. There is, however, a limit to the feed water preheating as soon as one approaches the temperature of the exhaust gases, which, in the case of good boiler plants, is somewhere about 180-200 degrees C. (356-392 degrees F.). It is no use to raise the feed water preheating still higher because in that case it is impossible for any part of the boiler surface to dissipate any heat of the fumes down to the desired temperature. Even if one approaches this limit, naturally extraordinary large heating surfaces are required for the cooling of the heating gases. For the same reasons it is no solution to set the air preheaters the task of dissipating the heat.

It is understood that the circumstances are more favorable if high pressure is used in plants with large units of, for instance, 1200 H.P. per shaft. At the relatively large quantity of steam the blades of the high-pressure turbine become necessarily longer, and therefore the clearance losses are decreased. There can be no doubt that in the moment in which one will have learnt

* With the use of fuel oil these figures are 0.308 kilograms, or 0.28 per B.H.P.-hour or I.H.P.-hour respectively.

to manufacture all elements from which a high-pressure plant is composed in a form which gives absolute safety in service, it will primarily be the field of large outputs in which the introduction of high-pressure steam finds its justification. But especially in this field it will be necessary to extend greatest care, as it always comes into question with high-class passenger ships. As mentioned above, the only solution possible in this case would be to design part of the plant as high-pressure boilers, but to work the rest at pressures within the normal limits (when using water-tube boilers 30 atmospheres are understood to be within these limits up till today), and to introduce the super-pressure steam into an additional turbine.

We come to the conclusion that from the standpoint of the marine engineer the support of experiments with super-pressure by installation of experimental boilers is always welcome in the interest of technical progress. On the other hand, it will be necessary to limit these experiments to the extent that only a small part of the boiler plant is to be equipped for super-pressure, while the principal part works at pressures which, under the present circumstances, are to be considered as normal.—"The Marine Engineer and Motorship Builder," June, 1931.

THE RESISTANCE OF COPPER-NICKEL STEELS TO SEA ACTION.*

By J. Newton Friend, D.Sc., and W. West.

It has long been known that the addition of upwards of 3 per cent of nickel to ordinary carbon steels enhances their resistance to corrosion, and it has recently been demonstrated in a considerable number of field tests in various parts of the world that small amounts of copper have an analogous effect. It seemed of interest to inquire, therefore, what the combined influence of these elements might be, and the following research was undertaken several years ago for the authors' own information. At the request of several friends to whom the results have been communicated, it was decided to offer the data for publication, in the hope that they might prove useful to others.

The steels were prepared in lots of 56 pounds. Metallic nickel and copper were heated in graphite crucible pots over a forced-draught coke fire with every precaution to avoid oxidation. To these metals definite quantities of liquid steel were added from a basic-lined electric furnace; the contents of the pots were well stirred, and cast into sand-moulds. The casts were annealed at 920 degrees C. for two hours, and allowed to cool in the furnace, after which the excess metal was sawn off and discarded. There was no outstanding feature to be noted in the preparation of these alloys. Homogeneity was only possible in the liquid state by the use of very hot steel having an approximate temperature of 1600 degrees C.

The chemical composition of the bars is given in Table I.

Alloy LD3 proved to be unmachinable, even after annealing several times, and was not proceeded with further. Alloy LD7 proved to be brittle; one specimen cracked during the pasty stage of cooling, and two others cracked during handling in the vice. The mechanical properties of the alloys are recorded in Table II.

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* Paper presented at the Iron and Steel Institute Annual Meeting, London, May 7 and 8, 1931. Abridged.

† Iron and Steel Institute, Carnegie Scholarship Memoirs, vol. xvi, page 131 (1927).

TABLE I.—Chemical Composition of the Bars.

——	*LD1.*	*LD2.*	*LD4.*	*LD5.*	*LD6.*	*LD7.*	*LD8.*	*LD3.*
	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.
Carbon	0·28	0·20	0·16	0·18	0·23	0·21	0·25	0·18
Manganese	0·69	0·58	0·63	0·53	0·58	0·52	0·61	0·55
Silicon	0·28	0·25	0·33	0·33	0·24	0·30	0·44	0·20
Sulphur	0·03	0·02	0·022	0·036	0·036	0·018	0·026	0·028
Phosphorus	0·037	0·029	0·027	0·022	0·036	0·022	0·026	0·031
Nickel	3·75	3·44	0·26	0·26	0·40	3·80	0·34	3·90
Copper	1·16	2·35	1·13	2·44	3·70	9·84	0·00	3·58

TABLE III.—Results of Exposure to the Sea for Two Years, 1927 to 1929.

Bar.	Condition.	Composition— Cu.	Composition— Ni.	Loss in Weight.	Relative Loss in Weight for Equal Areas.	Depths of Deepest Pits— Sides.	Depths of Deepest Pits— Ends.	Remarks.
		Per cent.	Per cent.	Grm.		Mm.	Mm.	
LD8 (1)	Annealed ..	—	0·34	41·34	104 } 100	0·64, 0·28	0·45	Surfaces pinholed, otherwise good. Slight corrosion where bars entered frame
(2)	,, ..			36·25	96	0·32	0·50	
(F)	Forged ..			35·70	89·3	0·15, 0·54	0·84	
LD4 (1)	Annealed ..	1·13	0·26	38·73	100·1	—	0·95	Slight pitting on faces. Ends locally attacked.
(F)	Forged ..			28·88	108·1	—	—	Round bar, uniformly attacked.
LD5 (1)	Annealed ..	2·44	0·26	36·36	96·0	0·41, 0·51	0·44, 0·33	Sides and ends pitted.
(F)	Forged ..			39·25	93·3	0·47, 0·69	0·30	
LD6 (1)	Annealed ..	3·70	0·40	34·58	88·3	0·48, 0·23	0·51, 1·33	Localised corrosion on every side and at ends.
(F)	Forged ..			37·42	96·4	0·39, 0·40	1·05	
LD1 (1)	Annealed ..	1·16	3·75	20·33	52·4	—	0·61, 0·62	Excellent. Ends only attacked.
(F)	Forged ..			22·90	53·7	—	0·39, 0·87	Tendency on one face to pit Ends attacked.
LD2 (1)	Annealed ..	2·35	3·44	26·75	68·5	0·45	0·50, 0·78	Slight pitting on several and localised corrosion at ends.
(2)	,, ..			27·45	71·0	—	0·69, 0·50	
(F)	Forged ..			45·42	113·0	0·27	0·93, 0·32	
LD7 ..	Annealed ..	9·84	3·80	27·08	75·5	—	0·75, 0·85	Do.

Conditions of Exposure.—The bars, which measured 9 inches in length, and 1⅛ inches square in cross-section, were fixed into a pitch pine rectangular frame, similar to those used in earlier researches, and bolted at half-tide level to Birnbeck Pier, Weston-super-Mare, under the supervision of Mr. Ernest J. McKaig, whom the authors take this opportunity of thanking. Particulars of the currents and analyses of the sea-water have already been given.† In the earlier researches the ends of the bars were plugged into the wooden frame with putty, but this proved unsatisfactory, as severe localized corrosion resulted when the putty disintegrated. Soft pitch was used in the present research and proved highly satisfactory. The frame was exposed to sea action for two years—namely, September, 1927, till October, 1929—after which the bars were removed, scraped clean, and examined. The results are given in Table III.

TABLE II.—*Mechanical Properties of the Alloys.*

Bar.	Condition.	Yield Point. Tons per sq. in.	Maximum Stress. Tons per sq. in.	Elongation on 3 in. Per cent.	Reduction of Area. Per cent.
LD8	Annealed ..	24·0	36·6	18	24·0
	Forged ..	24·4	38·6	28	52·5
LD4	Annealed ..	24·6	32·7	26	40·0
	Forged ..	25·8	35·0	26	43·0
LD5	Annealed ..	28·4	36·6	17	28·0
	Forged ..	37·0	49·0	18	45·0
LD6	Annealed ..	—	24·0	3	4·5*
	Forged ..	38·8	47·4	20	41·5
LD1	Annealed ..	31·8	45·0	2	4·5*
	Forged ..	—	68·0	Broke before reaching yield point and outside gauge marks	
LD2	Annealed ..	39·6	51·0	15	20·0
	Forged ..	—	88·2	8	24·0
LD7 ..	Annealed ..	—	6·0*	—	—

* Unsound Bar.

Consideration of the Results.—In general, the forged alloys suffered slightly greater corrosion than the corresponding annealed bars. The presence of nickel greatly refines the structure of cast steel, but there is no evidence that copper does so likewise. The addition of copper, up to 3.70 per cent, increases the resistance of steel to alternate wet and dry sea action. The most resistant steel, LD1, contained 1.16 per cent of copper and 3.75 per cent of nickel. It lost least in weight, and its general surface was free from pitting. It is noteworthy that this alloy possessed a fine structure; but resistance to corrosion in these experiments does not altogether coincide with fineness of grain, otherwise the forged LD2 and annealed LD7 would have corresponded more closely with LD1. The increased corrosion of LD2 may be connected with its more heterogeneous structure.—" Engineering," June 12, 1931.

U. S. S. *AKRON*.

Flight trials of the *ZRS-4*, dirigible building at the Goodyear-Zeppelin Corporation plant, Akron, Ohio, and to be christened U. S. S. *Akron* August 8 by Mrs. Hoover, will commence sometime after August 15, with a Navy crew and under careful scrutiny of a Naval Board of Inspection and Survey.

During these trials, at least 75 hours' flying will be carried out in at least five separate flights. Speed trials will be conducted, including a deceleration test; turning circles will be measured; altitude trials will be conducted; dynamic lift will be measured; interior ventilation and pressure equalization will be studied; fuel consumption will be determined and deflections and strains within the structure will be measured. Pressure distribution over the hull will be checked and various other scientific data bearing on the general air-worthiness and suitability of the airship will be collected and carefully evaluated.

Throughout her construction, various parts of the *ZRS-4* and her equipment have been critically examined to prove their suitability. Step by step all parts have been tested. The first section of the airship to be completed, a section near amidships, was subjected to a proof test, using for this purpose an appropriate gas cell, inflated with helium so as to simulate actual conditions as closely as practicable. Various loadings were applied to this section and scientific measurements made. The behavior of this critical section of the airship was according to calculations. Hundreds of tests of smaller magnitude have been made on girders, joints, fittings, fins, rudders, and various items of installation and equipment. As the airship approaches completion, other ground tests will be made, including hogging and sagging tests, inclining tests, electrical bonding and insulation tests, testing of fuel lines, control systems, and other apparatus.

The design of the *Akron* represents a special effort to stress safety from every standpoint. Structural integrity, accessibility for repairs, insurance against breakdown of any essential operational feature, protection against fire, and means for extinguishing fire are some of the major safety features of the airship's design.

The history of the design of the Navy airship *ZRS-4* (U. S. S. *Akron*) really goes back to 1924. The *Shenandoah* had been completed and the *Los Angeles* had been delivered from Germany. There was no immediate prospect of building additional airships, but in order to keep abreast of progress and to be ready with a modern design when and if necessity arose, the Bureau of Aeronautics commenced preliminary work on the design of an airship of about six million cubic feet gas volume. This size was chosen because, figuring on helium for inflation, it gave an airship of dimensions that would fit into the Lakehurst shed; an airship that could cross and recross the Atlantic at cruising speeds without refueling; and an airship of performance which compared with the two British airships of five million cubic feet volume that were to be inflated with hydrogen. The work on this new design was the medium for detailed technical consideration of various hull shapes, types of girders, types of main frames, most advantageous spacing of frames, various arrangements of engines and gas cells, and other problems of airship design. The general design work was reinforced by wind tunnel tests on models and by full scale tests on the *Los Angeles*, as well as by studies and experimental tests on various elements, such as girders and materials.

After the loss of the *Shenandoah* in 1925, it was considered desirable to very markedly increase the strength requirements hitherto in use in airship design. In revising upwards the conditions an airship could meet, it was found desirable to increase slightly the size of this airship to about six and one-half million cubic feet and to make the strength of the airship such that she could withstand conditions two to three times as severe as those sustained by the *Shenandoah*.

In 1926, a bill was introduced in Congress by the then chairman of the Naval Affairs Committee, the late Representative Thomas M. Butler, to authorize the building of an airship to replace the *Shenandoah*. Hearings were started and the airship question gone into exhaustively. Airplane

questions were considered, as well, and as a result of these hearings, the so-called Five-Year Aviation Program Bill was brought out and passed. Included in this bill was authorization for construction of two rigid airships of about six and one-half million cubic feet volume to be suitable for use with the Fleet. Funds for starting construction of these airships were not immediately available, but finally did become available in 1927.

In the interest of encouraging the establishment in the United States of an airship industry, it was decided that bids should be called for through a design competition rather than to build the new airships in the Navy Department's own plants as had been done in the case of the *Shenandoah*. The design which was then under way in the Bureau of Aeronautics was used as the basis for the preparation of a set of technical specifications describing a type of airship suited to Naval purposes. They were purposely made broad so that they might be fulfilled by any one of several designs. These technical specifications were used in the two design competitions which preceded the award of contracts for two airships to the Goodyear-Zeppelin Corporation in October, 1928. These specifications have been adhered to very closely, although naturally there have been departures in details from these specifications as the Goodyear Zeppelin Corporation has proceeded with detailing the design submitted by them and accepted under the 1928 Airship Design Competition.

The salient characteristics of the U. S. S. *Akron* are:

Nominal gas volume (gas cells 95 per cent full) cubic feet	6,500,000
Total air volume displaced, cubic feet	7,400,000
Length overall, feet	785
Maximum diameter, feet	132.9
Width overall (propellers vertical) feet	137.5
Height overall, feet	146.5
Normal spacing of main frames, feet	74
Number of gas cells	12
Weight empty or dead weight (estimated) (See Note 1), pounds	221,000
Useful lift (estimated) (See Note 2), pounds	182,000
Total lift (based on nominal gas volume and helium lifting .062 pounds per cubic foot), pounds	403,000
Ratio useful lift per total lift	.452
Number of engines	8
Maximum horsepower (sea level)	4480
Maximum speed (at 3000 feet altitude) (estimated) knots	72.8 (84.0 m.p.h.)
Normal fuel capacity (approximately 70 per cent of useful lift) pounds	124,000

Speed	Estimated Still Air Range	Estimated Hourly Fuel Consumption
Knots	Nautical Miles	Pounds
72.8	4800	1875
60	6600	1150
50	9000	700
40	13,000	380

Crew complement { Men 77 / Officers 12 (plus plane pilots)

Normal flying crew { Men 38 / Officers 10 (plus plane pilots)

Note 1. Weight empty (subject to correction for probable overweight) includes the complete hull structure, power plant and accessories, outer cover, ballast installation, landing and handling equipment, instruments and miscellaneous items permanently affixed to the airship.

Note 2. Useful lift includes crew and effects, food and fresh water, water ballast, fuel, oil, reserve cooling water, spare parts, military equipment.

The general appearance of the *Akron* is similar to the *Los Angeles* and other airships of the type. Some of her outstanding features are:

A fatter shape, her length being 5.9 times her maximum diameter.

Rather large stabilizing fins.

A single, rather small, protruding control car.

Eight internal engine rooms, which eliminate the pendant external power cars heretofore customary.

Eight propellers suspended from outrigger struts with means for tilting each propeller, so as to give thrust in four directions.

An internal hangar for housing four or five airplanes.

Resilient bulkheads for checking surging of the gas cells in case one is deflated.

Use of combined automatic and manually-operated gas valves.

Provision of special points at the bow for mooring the airship and under the lower fin for supporting the stern of the airship during ground handling operations.

Easier and better access to all parts of the interior of the airship.

Provision of machine gun emplacements.

The airship is constructed on the orthodox multi-layer principles which have applied to rigid airship construction, particularly of the Zeppelin type, during the past quarter of a century. The basic hull structure is a wire-braced system composed of transverse frames connected by longitudinal girders. The number of longitudinals is 36, except near the ends of the airship. In general, three intermediate frames are located between each two main frames. Three longitudinal corridors, or gangways, one on the top center line and two in the lower part of the hull, give the structure additional ruggedness. The engine rooms are located along the lower corridors. The gas cells, 12 in number, are placed inside of this hull structure, one gas cell being located between each pair of main frames. The gas cells are enclosed in a system of wire and cord nettings which serve to transmit the gas' forces to the hull framework. A taut outer cover of cotton cloth is applied and given several coats of acetate dope to protect the entire hull structure against atmospheric influences and to offer a smooth hull contour. Thus, if we start from the exterior and peel off successive layers, we would remove, first, the outer cover; second, the hull structure; third, gas cell nettings; fourth, the gas cells themselves, constructed of cotton cloth made impermeable to passage of gas; and finally, the helium gas itself which gives the airship its lifting power.

The power plant selected for the *Akron* comprises eight Maybach model *VL-2* gasoline engines, which type is a development from the 420-horsepower Maybach engines installed in the *Los Angeles* in 1924. The *VL-2* engine carrries a sea-level rating of 560 horsepower at 1600 R.P.M., with a compression ratio of 7 to 1. The engine is a 12-cylinder, 60 degree V-design, water-cooled, with aluminum pistons and crank case. All bearings are anti-friction, the main and connecting rod bearings being of the roller type. Compressed air is used for starting. Reversibility is obtained within the engine itself by having an ahead and an astern set of cams on the cam shaft which may be shifted longitudinally to bring either set into operation. The weight of this engine, dry, is approximately 4.5 pounds per horsepower. Fuel consumption is about .45 pound per horsepower of power.

Each of the eight engine rooms on the *Akron* has a floor space of about 8 × 8 feet. The engines are mounted athwartships and each engine drives its own propeller, through a 16-foot transverse shaft, which carries at its outboard end a tiltable housing from which the short propeller shaft protrudes. The short propeller shaft, normally horizontal, may be swung downwards through 90 degrees in a few seconds by means of a hand wheel located in the engine room. This feature, coupled with the reversibility of the engine, permits thrust to be obtained in four directions—ahead, astern, upwards, and downwards. An upward thrust of about 6000 pounds and a downward thrust of about 10,000 pounds are expected to be of assistance in mooring and handling the airship. When the airship is in normal flight, it will be more economical to obtain vertical movement of the airship through the use of rudders rather than resort to the tilting propellers.

Each engine room contains the usual engine instruments, oil service tanks, and a specially built blower to supply fresh air for intake purposes, crankcase ventilation and ventilation of the engine room itself. Signals which guide the engine mechanic in controlling the engines are received from the control car by a mechanical telegraph or indicator. Gasoline storage tanks are located in the corridor nearby and feed by gravity to the engine carburetor. The engine rooms are very carefully fireproofed and each is provided with an automatic carbon dioxide fire extinguisher in addition to hand type fire extinguishers and other fire protection apparatus located in the engine rooms and immediately adjacent thereto.

The transmission gearing which connects each engine with its propeller includes a speed reduction of .578 so that 1600 R.P.M. at the engine crankshaft are reduced to 925 R.P.M. at the propeller. The weight of this transmission gearing is approximately 1600 pounds, and its mechanical efficiency is estimated at 97 per cent or better. This transmission gearing has been subjected to a very careful endurance test of more than 300 hours, during which its torsional characteristics have been thoroughly investigated with the result that any possible dangerous peak of torsional vibration is believed to have been located and placed outside the operating range of the power plant.

The two-bladed, wooden tractor propellers are about 16 feet 4 inches in diameter, and are designed to take into account the effect on their efficiency of the multiple tandem arrangement which results from having four propellers on each side located approximately in the same horizontal line and 74 feet apart.

Engine radiators will be of the ordinary core type mounted on the outriggers which carry the transmission shaft. To give some measure of cooling when propellers are acting vertically, supplementary cooling arrangements in the form of horizontal finned tubes have been provided.

The usual method employed in the United States to preserve equilibrium of a helium-filled airship, and thereby to avoid the necessity for valving buoyant gas as fuel is consumed and the airship tends to become light, is to employ a water recovery apparatus which condenses the moisture content of the engine exhaust to water and retains the water so recovered on board the airship as ballast. Theoretically, it is possible to recover 135 pounds of water in this manner for every 100 pounds of aviation gasoline that is burned, but this figure will vary according to several factors. The water recovery apparatus on the U. S. S. *Akron* represents an effort to install the apparatus close to the hull of the airship, thereby making it somewhat of a skin type condenser and so reduce its drag or resistance. The apparatus comprises five panels mounted close to the hull above each engine. Each panel consists of horizontal aluminum tubes connected by vertical headers. The flow of gas initially is upwards and the condensed water is drawn off through by-pass pipes and circulated to fabric bags throughout the airship.

The normal gasoline supply of about 124,000 pounds is stored in a total of 110 aluminum tanks of three sizes, the majority of them being of 120 gallons capacity. These tanks are located alongside the two side corridors, convenient to the engine rooms, and are so arranged as to properly trim the airship. A rather extensive system of piping, principally aluminum, permits fuel to be received at the bow or near amidships and to be circulated at will. Fuel also may be shifted in flight from one container to any other container. Oil is stowed in eight 1500-pounds capacity tanks, one being located near each engine room.

The ballast system comprises some 44 rubber fabric storage bags of several sizes connected by a system of piping. Each of these bags is equipped with a quick discharge valve which can be operated through a wire pull leading to the control car. Certain of these ballast bags are located near the bow and stern of the airship and serve especially as emergency ballast and for correcting the trim of the airship. The remaining bags are located along the length of the airship and may be used in the same way for emergency purposes, but their major purpose is to serve as storage bags for recovered ballast water.

Each gas cell is a fabric cylinder dimensioned to fit its particular location in the airship. The largest cell is near amidships and has a capacity of some 980,000 cubic feet of helium. It is 74 feet long and approximately 130 feet in diameter. One-half of the cells in the *Akron* will be made of 2-ounce cotton cloth coated with several layers of rubber and a final coat of paraffin. The other half of the *Akron's* cells will be made of an improved and somewhat more expensive fabric which is made up from 2-ounce cloth coated with several layers of a mixture of gelatin and rubber latex, with a final coat of paraffin. The final weight of gas cell fabric is between 5.5 and 6.5 ounces per square yard. The total yardage of cloth in the gas cells is about 56,000.

Gas cells are located along the upper corridor and are either fully automatic or a combined automatic and manually-operated type. The manual operation is by means of wire pulls leading to the valve control switchboard in the control car. The valves are 32 inches in diameter and the number of valves is arranged so that the airship may rise at a rate of 4000 feet per minute without causing serious increase in internal pressure. Adequate openings are provided in the hull covering so that pressure inside the hull may equalize quickly as the airship ascends or descends. In the plane of each of the main frames there is a slack bulkhead of hard wires which looks very much like a spider web. These bulkheads are located between the gas cells and are fitted with a resiliency device (gas-filled cylinders) so that some bulging of the bulkhead may take place in the event a gas cell pushes against it, and serious torsional loads in the main frames will not thereby be built up. In order to take care of possible bulging of gas cells, extra material has been provided in the circular ends of the gas cells abreast the resiliency device.

The outer cover of the *Akron* is made up from 2.8-ounce cotton cloth, sewed into panels approximately 74 feet long by either 12 or 24 feet wide, bounded by an eyelet lacing edge. These panels are laced into position and secured to the hull framework and to special supporting wires. The panels are then given four coats of acetate dope, the last two coats of which contain aluminum powder. The final weight of the outer cover is approximately 5.3 ounces per square yard, and a total yardage of about 36,000 square yards is required.

The *Akron* contains an electrical plant and system which is substantially a miniature reproduction of similar systems used on destroyers, submarines and other vessels. Power is needed for radio, lighting, telephones, a portion of the cooking, and for certain pumps, winches, fans, etc. The main source of power is two 8 Kw., 110 V., D. C. generators, each driven by an independent gasoline engine. These two generators may be paralleled in opera-

tion. In addition, there is a small reversible dynamotor and a 130-hour, 24-volt storage battery. All of this equipment is installed in a fireproof generator room located immediately adjacent to one of the forward engine rooms.

The radio outfit was built to specifications developed by the Bureau of Engineering. It comprises an intermediate frequency transmitter (300-605 kilocycles) and a high frequency transmitter (3000-18,100 kilocycles) and is expected to show a transmitting range on high frequency of at least 5000 nautical miles, and on intermediate frequency of at least 500 nautical miles. Approximately 500 watts' antenna input is contemplated for each transmitter. In addition, there is a light-weight radio compass. Facsimile equipment will be installed later. Trailing wire types of antenna are arranged to be reeled in by specially-built electrical winches. In addition, there is a short fixed-wire receiving antenna attached to the hull of the airship. Duplicate motor generators are mounted just outside the radio room and supply high voltage current required for the radio outfit.

An 18-instrument telephone system is provided with a master switchboard located in the control car. Three conversations may take place simultaneously over this system, or the switchboard operator may call at least three outlying stations simultaneously, or he may sound all telephone alarms, using code if desired.

The control car or room is located near the forward part of the ship so as to obtain the best possible vision for operating purposes. The forward third of this control car is the location for rudder and elevator controls, gas and ballast controls, instruments and other apparatus which is essential to the functioning of the airship. The middle third of this control car is the navigating compartment, and the after portion of the control car is for access hatches and ladders. An emergency control station is located in the lower fin. Underneath the control car and the lower fin provision is made for detachable rattan bumpers. Above the control room are the radio and aerological rooms, quarters for the captain and a part of the officers; and also an office space and photographic laboratory.

Living accommodations are concentrated nearer amidships, abreast the airplane hangar compartment. On the port side there is a crew's toilet and wash room and seven rooms, each having a floor area of about 8 × 10 feet, and each being fitted with four canvas-bottom bunks and locker spaces. On the starboard side, from aft forward, there are the generator room, galley, crew's mess, chief petty officers' mess, officers' mess, and two four-bunk rooms for officers. In order to save weight, it is customary in airships for more than one man to use in rotation the same bunk, and this will be done in the U. S. S. *Akron*. The normal flight crew of the *Akron* will probably be 38 men and 11 officers, plus whatever personnel is required to operate airplanes. Along the corridor abreast the living spaces, cellon windows are arranged, but because of the contour of the hull in this region, these windows are nearly horizontal.

In the galley there is a propane gas stove and hot water heater, and these are supplemented by auxiliary electrical apparatus for serving coffee and night rations, making it unnecessary to light off the main gas-fired equipment except to prepare a full meal. The galley is fireproofed and is fitted with rather elaborate arrangements for taking care of garbage until it can be dumped overboard with safety. Provisions will be carried in conveniently arranged storerooms and some cold storage arrangements will be added. Drinking water is carried in special tanks filled through an independent piping system.

For practically the first time in airship construction, effort has been made to provide heat for living spaces. Warm air, after passing over engine manifolds in the forward engine rooms, is led to a large insulated aluminum

pipe which runs under the floors of the living spaces and forward to the control car spaces. Registers in the floors of the living spaces which may be opened or closed give the possibility of regulating the heat.

The airplane compartment is a specially arranged space about 70 feet long by 58 feet wide by 16 feet high. In order to provide entrance and exit for the planes a T-shaped hatch 32 feet wide by about 40 feet long is provided and may be closed by part-folding, part-sliding doors. It was necessary to provide openings in several longitudinals for this hatch; continuity of strength, however, is secured by deep framing above the compartment. This overhead framing will carry the tracks on which the airplanes will move to their storage positions—one in each corner and a fifth in the center. The contractor will deliver the airship with the airplane compartment bare, and the Navy Department is to supply the necessary trapeze or other device for landing and launching the airplanes, stowage arrangements, and the airplanes themselves. Some months will probably elapse before a final arrangement of trapeze, trolleys, etc., is determined upon and installed. For the first experiments, and until types of airplanes more suited to airship uses become available, probably only four airplanes will be carried at any one time.

Seven gun emplacements have been located so as to cover all angles of approach and some of these emplacements will carry more than one gun. The final choice of guns, their arrangement, and means for ammunition stowage and supply are matters that will require to be worked out on the basis of actual experience, including joint maneuvers between airplanes and airships.

An observation basket which may be lowered several hundred feet below the airship is contemplated for future installation.

The *Akron* carries at her bow the conventional type of horizontal mooring spindle and pendant cone, the cone being dimensioned according to generally accepted international standards. The lower fin of the *Akron* is made especially strong to serve as a point of support during certain ground handling maneuvers. Various attachment points are provided on the hull in the region of the stern and elsewhere to which handling lines for mechanical handling systems, or manually-controlled lines, may be attached. Inside the airship, working platforms, winches and trap doors near the bow provide the means for handling the ⅞-inch main mooring cable and the two ¾-inch side or yaw cables used in mooring operations. In addition, there is a large door near the bow of the airship which may be lowered to form a bridge connecting the airship with the upper platform of the mooring mast and so provide a means of entrance and exit via the mast.—"Navy Press."

BOOK REVIEWS.

UNIFLOW, BACK-PRESSURE AND STEAM EXTRACTION ENGINES. By Eng. Lieut.-Com. T. Allen, R. N. (S. R.). Published by Isaac Pitman & Sons, 2 West 45th St., New York. Price $10.00.

"Uniflow, Back-Pressure and Steam Extraction Engines" is a very complete and comprehensive treatise for designers, practical engineers and students interested particularly in these three types of reciprocating engines. It contains many curves and tables, with their derivations, dealing with mean effective pressures, steam consumption and correct proportioning of the various parts of which the completed engine is composed, based upon a standardization system and in accordance with the best present day practice. There are examples, with detailed calculations of the design of each type of engine to suit existing working conditions. There are illustrations and descriptions of engines developed by well-known manufacturers, together with the practical use of each of the three types. The book also deals with the conversion of existing condensing plants of older design to Back-pressure or Steam Extraction engines and the necessary changes involved. Speed and pressure governors and their application to various governing systems are also discussed.

W. B. Newton.

THE WAR IN THE AIR. Vol. 3. (Official History of the War). By H. A. Jones.

The subject matter of this volume is best described by the first paragraph of the preface:

"This third volume tells the part played by aircraft in the destruction of the *Königsberg,* and in the campaigns against German East Africa and German Southwest Africa; gives an account of the air attacks on Great Britain, 1914-16; reviews the

problems of supply, administration, recruitment, and training, 1914-16, and outlines the air developments of operations on the western front in the winter of 1916-17, and during the battle of Arras, 1917."

The compiler has succeeded in arranging a mass of facts and incidents in a manner which makes interesting, instructive reading. His portrayal of the general political, ground, or strategical situation leads one up to the various air operations in an enlightening and clarifying manner.

Chapter I covers flying operations in German East Africa and Southwest Africa. One of the outstanding points of interest in this chapter is the description of the use of two 90-horsepower Curtiss flying boats, civilian owned, and civilian operated, in the search for the *Königsberg*. Needless to say the personnel operating the plane soon were inducted into the British Naval Flying Service. There is a very entertaining account of the earliest use of aircraft in the African area of operations, together with graphic description of early operating conditions under high temperatures and tropical moisture at the equator.

The operations against the *Königsberg* are well worth reading for the sheer enjoyment of the accounts of high adventure, persistence in the face of seemingly insurmountable difficulties and final success of the operations. The *Königsberg* was found in the Rufiji Delta late in October, 1914, and she was finally destroyed 11 July, 1915.

An excellent picture is presented of the weather and flying conditions in Africa, the disappointment of supply and equipment, the manner in which aircraft stood up under the operating conditions and the hardships suffered by personnel in all branches of the service. The flying conditions may be imagined from the account of one pilot who upon investigating a "suspicious-looking dust cloud" found himself engulfed in a cloud of migrating locusts! At times this chapter becomes rather dull when the author is describing the campaign on the ground, but the chapter is enlivened by accounts of the moral effect of airplanes and bombs upon the African natives, the stories of pilots and observers forced down in the equatorial area, and by following the ever growing demand of the ground forces for reconnaissance from

the air before, during and after each operation. With the planes available, there was little material damage from bombing, but the moral effect of this bombing completely warranted the operations. The planes were invaluable in reconnoitering terrain which ground troops probably could not have reconnoitered at all.

Chapters II and III cover the air raids upon Great Britain 1914 to December, 1916. It opens with a short discussion of international law as applied to bombardment from the air of "undefended" places and also describes the efforts made to protect England from aerial attacks. This is followed by an interesting discussion of coast defense (pages 73-74). Next there is a brief but complete survey of the lighter-than-air situation in Germany at the outbreak of the war. One is struck by the coincidence between the unreadiness of Great Britain's air defenses and the unreadiness of German airships for raids. The impression is gathered that, fortunately, Germany's development and acquisition of airships was closely paralleled by Great Britain's acquisition and development of defenses against aerial raids.

The compiler's description of the expansion of Great Britain's air defenses is well worth reading by anyone interested in problems of national defense.

The author takes a very lenient attitude toward the bombing of "undefended" points and toward the "intent" of the Germans when their bombs landed on undefended towns and villages. Great pains were taken, apparently, to check the British tracks of raiding airships against the tracks reported by the Zeppelin commanding officers. This comparison shows the errors of airship navigation: Zeppelin commanders reported bombing towns that actually were miles off their course. Throughout this chapter, the historian takes a very chivalrous attitude toward the Zeppelin commanders and their actions.

The chapter describes the difficulties encountered in getting civilian cooperation in the darkening of towns and describes the unexpected effectiveness of darkening towns to prevent their discovery by the raiding airships. It is interesting to note the entire inadequacy of the airplanes at first assigned to drive off the raiders, and the lack of training of the pilots of these planes. If the raiding airships came in high, the early planes could not attain

sufficient altitude to engage the raiders, and, added to this difficulty, the facilities and training for night flying were so limited and the casualties to planes taking off at night to drive off raiders was so high that the Admiralty considered prohibiting the planes taking off at night to attack the raiders. "Pilots, with few exceptions, had not the aircraft that would take them near or above the Zeppelins."

The historian touches on German airplane raids on Great Britain but these were so few that little space is given it.

The chapter describes graphically the vacillating attitude of the British authorities as to the responsibility for the defenses of Great Britain from air raids and, on page 157, it quotes the final decision as to the responsibility for air defenses. This is followed by an instructive and comprehensive description of the air defense system of Great Britain.

The chapter leads up in an entertaining manner to the ultimate success of anti-aircraft guns and airplanes in driving off or destroying German airships. The accounts of the various pilots who destroyed airships make fascinating reading. The author states that the German investment in Zeppelins was well worth while for raiding alone and not even considering their value to the fleet as scouts. He states that the raids caused diversion from the theater of war of a formidable organization comprising 17,341 officers and men and 12 R. F. C. squadrons comprising 2200 officers and men and 110 airplanes. The author outlines, in addition, the loss in output of war materials by the suspension of industrial activities preceding, during and just following air raids.

This chapter is interesting to Heaver-than-Air pilots as well as to Lighter-than-Air pilots. One is particularly impressed by the courage and determination of the German airship commanders in taking off and pushing home their attacks in the face of very adverse weather conditions.

Chapter IV covers administration, supply, recruitment, and training 1914-16. In this chapter the author deals with the efforts, political and industrial, to obtain satisfactory airplanes and reliable engines. He traces the history of the various boards, commissions, etc., for procurement of these materials. The chapter closes with a summary of the difficulties encountered in obtaining planes, engines, and personnel for the air arm.

Chapters V and VI cover the general situation on the western front during the winter of 1916-17 and the battles of Arras 1917. The summary of the western front situation is brief and clear. It prepares one to understand the air activities in the battles of Arras.

The author outlines the organization of the air force for controlling artillery fire, for intelligence service within the squadrons and for air fighting. He outlines the difficulties of training sufficient pilots and the lessons learned in filling squadrons with pilots for transfer to France. There follows a vivid description of the battles of Arras, 1917, which impresses one with the fine morale displayed by British pilots in the face of German superiority in numbers of planes, in the performance of planes, and in trained and experienced pilots.

The appendix contains splendid tables for persons interested in the material supplied the air force during the war, and to those interested in the characteristics and history of each German airship. Tables also summarize the date, result and extent of each air raid on England and Scotland during the period of this volume. The appendix also contains a brief discussion of the development of tracer, incendiary and explosive ammunition. Two most interesting documents are reprinted; these are: "Notes on Airplane Fighting in Single Seater Scouts," issued by the General Staff, November, 1916, and "Fighting in the Air," issued by the General Staff 1917.

The volume is of interest primarily to historians and to professional military personnel, but there is sufficient discussion of national defense problems to make the volume of interest to statesmen and framers of national defense policies. To the laymen the volume is interesting in the accounts of various air fights and in the discussions of cooperation between civilian and military personnel in modern warfare.

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JAHRBUCH DER SCHIFFBAUTECHNISCHEN GESELLSCHAFT (YEARBOOK OF THE GERMAN SHIPBUILDERS SOCIETY) 32ND VOLUME 1931. PUBLISHED FOR THE SOCIETY BY DEUTSCHE VERLAGSWERKE STRAUSS, VETTER & CO., BERLIN SW68. 436 PAGES, PRICE RM 20 (ABOUT $5.00).

This year's edition of the well-known blue-covered yearbook contains in addition to the customary list of members, necrologies, and the reports of the Society's activities, the following complete papers accompanied by the discussion which they provoked:

The Airship *Graf Zeppelin*, by E. Lehmann.
Building of Heavy Gun Emplacements, by H. Methling.
Wake and Airfoil Propellers, by H. M. Weitbrecht.
Wake and Airfoil Propellers, by G. Kempf.
Theory of Elasticity and Experiment, by G. Schnadel.
Theory of the Mast Stresses, by E. Pophanken.
Newer Ideas on Design of Ships' Rudders, by W. Kucharski.
High-pressure Marine Steam Engine, by J. Stumpf.
Freeboard and Measurement of Freighters, by R. Erbach.
Freeboard and Ship Measurement, by N. G. Nilsson.
Modern Problems of German River Traffic, by W. Teubert.
Methods of Manufacturing River Craft, by E. Ilgenstein.
River Shipbuilding and River Traffic, by Frentzen.

A Description of Excursions to the Prussian Experiment Station for Hydraulics and Shipbuilding, the German Experiment Station for Aeronautics, the Western Port of Berlin and a complete list of books in the Society's Library complete the volume.

E. C. MAGDEBURGER.

ASSOCIATION NOTES.

Due to detachment from duty in Washington, the Society has accepted with genuine regret the resignation of Rear Admiral H. E. Yarnell, U. S. N., as President, and of Rear Admiral John Halligan, Jr., U. S. N., and Captain I. E. Bass, U. S. N., as members of the Council. Rear Admiral S. M. Robinson, U. S. N., Chief of Bureau of Engineering, has been named President and Rear Admiral George C. Day, U. S. N., and Captain H. G. Bowen, U. S. N., members of the Council for the unexpired terms of the former officers.

MEMBERSHIP.

The following have joined the Society since the publication of the last previous JOURNAL:

NAVAL.

Verbrugge, Louis, Chief Machinist, U. S. Navy.

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